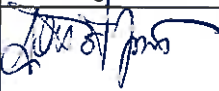
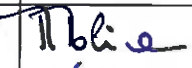

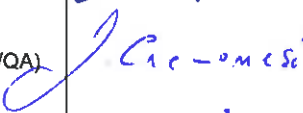





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1. SCOPE

This document provides the thermal model description and the thermal analysis results of the Cryocoolers Thermal Control System (TCS), whose design description is provided in [RD5].

This document corresponds to contract deliverable DEL 076.

2. APPLICABLE AND REFERENCE DOCUMENTS

Documents here below identified are applicable and/or reference for the activities described in the present document and are considered part of it to the extent specified herein.

2.1 APPLICABLE DOCUMENTS

CONTRACTUAL

AD1 Capitolato generale ASI, available on http://www.asi.it/html/norme/cap_gen.pdf

AD2 Richiesta d'offerta per Programma AMS, attività di Fase C/D – Prot. ASI 006194 – 25/07/2007

AD2bis Capitolato Tecnico "Progetto: AMS Attività di fase C/D" Doc. N. DC-IPC-2007-062

AD3 Tailoring di primo livello delle norme ECSS, serie M-E-Q – Progetto AMS attività di fase C/D- Doc. n° DC-IPC-2007-063 Rev. A

MANAGEMENT

AD4 "ECSS Glossary" – Doc. ECSS-P-001 Rev. B

PRODUCT ASSURANCE

AD5 "Product Assurance Requirements - Progetto AMS attività di fase C/D "-Doc. n° DC-IPC-2007-064 Rev. A


AD6 "Istruzione Operativa "Norme per la redazione del Piano di Assicurazione del Prodotto (PA Plan)", Doc. OP-IPC-2005-008

AD7 "Sistemi di Gestione per la Qualità", doc. UNI EN ISO 9001:2000

AD8 "Quality Management Plan for the Alpha Magnetic Spectrometer 02 (AMS-02) Experiment", Doc. JSC63164, Basic Version, 09/21/2005

AD9 "Master Verification Plan (MVP)", Doc. JSC 29788, Iss. Draft, 8/21/2006

AD10 "PA REQUIREMENTS DC-IPC-2007-064 RevA Conformity", doc AMSCD-RQ-CGS-001 issue 1


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- AD11** "Multi-Layer Insulation for the Alpha Magnetic Spectrometer Guidelines", Doc. CTSD-SH-1782, 9/30/2005
- AD12** "AMS-02 Structural Verification Plan for the Space Transportation System and the International Space Station", Doc. JSC28792, Iss.D, March 2005
- AD13** "Experiment/Payload Integration Hardware Interfaces - Part I", Doc. JSC29095, Iss.A, 06/01/2002
- AD14** "Experiment/Payload Integration Hardware Interfaces - Part II", Doc. JSC29095, Iss.A, August 2004
- AD15** "Experiment/Vacuum Case Payload Integration Hardware Interface", Doc. JSC29202, Iss.C, March 2005
- AD16** "AMS-02 Thermal Control Requirements", Doc. AMS-RQ-CGS-001, Iss.1, April 2007
- AD17** "Attached Payload Interface Requirements Document", Doc. SSP 57003, Iss. B, 17/06/03
- AD18** "Attached Payload Hardware Interface Control Document, Doc. SSP 57004, Iss. B, 13/06/03
- AD19** "AMSPDS-RP-CGS-001", Doc. PDS Design Description, Iss.2, July 05
- AD20** "AmsE-PPL", AMS Electronics Preferred Parts List, available on
(<http://ams.cern.ch/AMS/Electronics/Parts/>), Iss.1, Nov 01 configured on doc PDS-LI-CGS-006 iss 1
- AD21** "PDS ICD - Interface Control Document", Doc. AMSPDS-IC-CGS-001, Iss.3, July 05
- AD22** "PDS Specifications", Doc. AMS-RQ-CGS-002, Iss.1, April 07
- AD23** "Cryo TCS and Support Beams Specifications", Doc. AMSTCS-SP-CGS-011, Iss.1, 20/03/2008

2.2 REFERENCE DOCUMENTS

- [DR 1]** Phase II Flight Safety Data Package for the Alpha Magnetic Spectrometer - 02 (AMS-02) Version Basic JSC49978, 2006
- [DR 2]** Alpha Magnetic Spectrometer – 02 Assembly and Testing Integration Plan, Version A, JSC63123, 28-11-2005
- [DR 3]** Dichiarazione INFN sulla consegna di componenti – lettera del 20 maggio 2007 (prot. ASI n. 0009869)
- [DR 4]** Capitolato gestionale ASI OP-IPC-2005-010-E
- [DR 5]** AMSTCS-TN-CGS-013, *Cryo TCS and support beams Design Report*, 31/03/2007, issue 1
- [DR 6]** AMSTCS-IC-CGS-002, *Cryo TCS and support beams ICD*, 31/03/2007, issue 1

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3. MODEL HIERARCHY AND ASSUMPTIONS

AMS-02 is a complex system, whose detailed thermal modelling cannot be developed within a single model, due to the high detail level needed on each subsystem.

Therefore, at least 2 model levels exist:

1. a System level model, which includes a reduced model of each subsystem. This model includes AMS and the International Space Station (ISS). This model is used for orbital cases survey and for generation of interface data (see below)
2. several subsystem level detailed models, which contain the detailed description of each subsystem only. In order to provide results, these systems need as input the interface data provided by the system level model.

Interface data is the way the system model communicates to the subsystem detailed models. While the system model considers the mutual interactions of the subsystems, and the interactions with the ISS and the orbital environment, the subsystem detailed model receive as input a simplified and representative set of environmental conditions.

The interface data include

- the Mean Effective Radiative Temperatures (MERAT), which represent the average sink temperatures which each component of the subsystem sees
- the GR radiative couplings of the subsystem nodes to their MERAT
- the orbital fluxes impinging on the subsystem elements
- the interfaces elements temperatures, which are treated as boundaries temperatures in the subsystem detailed model.

These inputs allow a much detailed modelling of each subsystem, and temperature predictions with fine detail level on several different operational modes.

The Cryocooler TCS follows this concept: the system model contains a simplified version of the cryocooler TCS, with a reduced number of elements and simplified assumptions. This is used to generate the interface data which are used for detailed temperature predictions.

The detailed thermal model is a 2-phase model, where the LHP are modelled in detail and the nodalization is much finer.

The interfaces of the thermal model are constituted by the Zenith radiator outer skin MERAT, fluxes and GR. The cryocooler is considered as an interface node to which the dissipated power is applied, not linked with conductive paths to the Vacuum Case. All the cryocooler power is assumed to be managed by the LHP system (for justification see section 5.1.2)


The transport lines are considered adiabatic, being wrapped by MLI, and therefore no interface data are provided for them (see section 6.1).

The zenith radiator lower skin is considered adiabatic (due to the MLI and spokes design, see for justification section 6.2).

The radiator thermal model at system level, which was used for the generation of the interface data for the 2-phase modelling, is characterized by the following properties:

Radiator property:	BOL	EOL
Area (per quadrant)	1.2m ²	
Emissivity (epsilon)	0.78	0.75
Absorptivity (alpha)	0.08	0.13

Tab. 3-1: Silver Teflon 5 mil optical properties)

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A survey on the orbital cases has led to the definition of the following dimensioning cases, used in the analysis reported in the following sections:

Orbital case:	Beta angle	Attitude (Y,P,R)
Worst hot case	+75°	+15+25-15
Worst cold case	-75°	-15-20-15

Tab. 3-2: Silver Teflon 5 mil optical properties)

The environmental parameters of the hot cases are characterized by a high solar constant (1424W/m²), a high Earth radiating temperature (266.5K) and high albedo (0.4)

The environmental parameters of the cold cases are characterized by a low solar constant (1310W/m²), a low Earth radiating temperature (244.5K) and low albedo (0.2).

4. 2-PHASE MODEL RESULTS

A 2-phase detailed thermal model has been built and run in the worst hot and worst cold orbital cases. The model has been built implementing the interfaces as per previous section (only radiator outer skin Interface data).

Thermal behavior has been predicted at three different power levels, corresponding to the minimum, nominal and maximum power that a cryocooler can dissipate, namely 63, 105 and 158W. It is worth recalling that 158W represent a failure mode, where a cryocooler is not working and the thermal load is shared by the three remaining units. Analyses have been carried on both the upper and lower cryocoolers lines, which differ in the Compensation Chamber size and in the transport lines length. Thermal performance differences are however negligible.

In all the 2-phase model simulations, the effects of 2 LHP's working simultaneously are considered. For LHP failures results, see section 5.1.1 of this document.

The fluidic model description, all the modeling parameters, and the thermal analysis results are provided in annex 1. In the following, the main results are summarized


Case	Power Level	Cryocooler Min/Max temperatures (°C)	Margin ¹ , °C w.r.t. -30/+40°C	Remarks
Hot case	63W (low)	-2/-1	+28/+41	Valve actuation → negligible oscillations
	105W (nominal)	+5/+10	+35/+30	Bypass valve is partially actuating
	158W (maximum)	+23/+30	+53/+10	
Cold case	63W (low)	-3/-3	+27/+43	Valve actuation → negligible oscillations
	105W (nominal)	+2/+2	+32/+38	Valve actuation → negligible oscillations
	158W (maximum)	+8/+8	+38/+32	Valve actuation → negligible oscillations

Tab. 4-1: Summary of temperature results at cryocooler interface (on cryocooler rejection collar)

Temperatures on the cryocooler body (where operational requirements in the range -30++40°C are set, see [AD23]) are fulfilled with large margin.

Upper and lower cryocoolers are behaving in a almost identical way. The bypass valve in cold and in low power cases is able to damp temperature oscillations (within 1°C), while granting a very stable temperature above the lower required level.

¹ Margins are evaluated as: [Tmin(calculated) – Tmin(requirement)], and [Tmax(requirement) – Tmax(calculated)]; positive margins therefore correspond to requirement fulfilment at both ends.

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5. OFF NOMINAL RESULTS

In the following section, off nominal results are presented in the cases of failures or non-nominal conditions:

1. Failure of a LHP
2. Failure of a Cryocooler
3. Failure of heaters thermostat in closed mode
4. AMS-02 experiment loss of power.

5.1 FAILURE MODES

5.1.1 1 LHP OFF

In the case of 1LHP failure, the system has to be able to manage the dissipated cryocooler heat with a single working LHP. In this case, the power to be considered is the nominal power, 105W. [the use of 158W input power is in fact representative of the failure of one cryocooler, and cannot be superimposed to a LHP failure].

The thermal behaviour of the TCS in case of LHP failure can be inferred according to the following consideration:

1. The radiator temperature is dependent only on the total amount of power, not on the number of LHP. Therefore, we can expect to have identical temperatures at radiator level.
2. The temperature drop between the LHP condenser and the radiator skin is instead doubled with respect to the nominal case when 2 LHP are working, since the double of the power is transferred through the same interface.
3. The temperature drop between the condenser and the evaporator shall double for the same reason.
4. Finally, the temperature drop between the evaporator and the cryocooler collar shall double as well, since the same total amount of power is now transferred through just the half of the interface area.

Basically, this means that the total temperature drop Radiator-Cryocooler needs to be doubled in order to calculate the performance with a single LHP working.

Temperatures and temperature drops of the nominal case (2LHP, 105W) can be taken from Annex1, and properly treated in order to calculate the LHP failure mode temperature effect on the cryocooler interface.

Nominal Case:

Radiator Maximum temperature: -6°C
Cryocooler Maximum temperature $+10^{\circ}\text{C}$
Total Drop Rad-Cryo: $+16^{\circ}\text{C}$

→ Total Drop, 1LHP = 32°C

→ Cryocooler max temperature, 1 LHP failure: $-6^{\circ}\text{C} + 32^{\circ}\text{C} = +26^{\circ}\text{C}$

The validity of this assumption is justified also thanks to the comparison of the 2 LHP cases at different power, assuming that the temperature drops are proportional to the heat amount:

- a. SINCE @ 105W → 16°C drop
- b. THEN @ 158W → $16^{\circ}\text{C} * 158/105 = 24^{\circ}\text{C}$ gradients are expected.
- c. being the max radiator temperature equal to $+7.5^{\circ}\text{C}$ (at 158W) → $7.5+24 = 31.5^{\circ}\text{C}$ cryocooler temperature is expected
- d. 30°C are derived from simulation → the proposed interpolation method has brought a (conservative) error of 1.5°C (conservative, in the sense that at higher power the LHP performance seems to generate proportionally lower gradients, thanks to the 2-phase non-linear heat transfer method).

The considerations above lead to a Cryocooler temperature which is 14°C lower than the maximum allowable value, in the single LHP failure case. This provided large enough margins to be confident of the robustness of the design.

According to this interpolation method, one can write:



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$$T_{Cryo} = T_{Rad} + 0.305 \frac{Q_{cryo}}{N}$$

Where Q is the power applied and N is the number of working LHP.
The radiator temperature is well interpolated by the relation:

$$T_{RAD} = \sqrt[4]{T_s^4 + \frac{Q_{cryo} + Q_{orb}}{GR}} - 273 = \sqrt[4]{109^4 + \frac{Q_{cryo}[W] + 145}{5.6E-8}} - 273$$

Hence:

$$T_{Cryo} = \sqrt[4]{109^4 + \frac{Q_{cryo}[W] + 145}{5.6E-8}} - 273 + 0.305 \frac{Q_{cryo}}{N}$$

The results are plotted in the following figure, as reference. Maximum theoretical power which can be dissipated by the cryocooler when 1LHP only is running is 130W (in order to maintain the temperature below 40°C). Please note that in the figure below the effect of the bypass valve are not taken into account. They produce a flattening of low temperatures around 0°C.

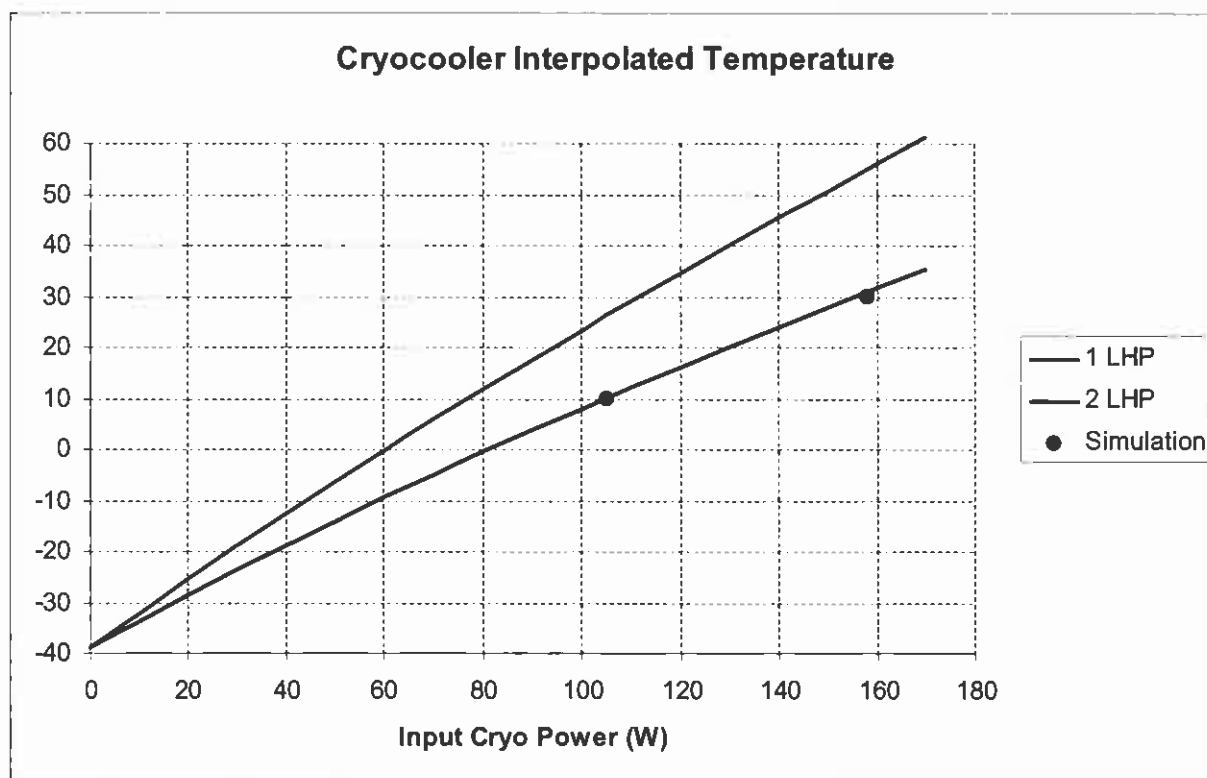



Fig. 5-1: Interpolated cryocooler temperatures as a function of the input power, in the 1 and 2LHP cases. Simulation results are represented by circles

5.1.2 1 (OUT OF 4) CRYOCOOLER OFF

The potential failure of a Cryocooler leads to a redistribution of the heat loads on the three remaining ones. According to the Cryocoolers providers, the total cooling capability of the entire system is maintained provided that the cryocoolers are operating at higher power, up to a total heat rejection of 158W each.

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Under these (failure) conditions, the operations of the TCS (with 2 LHP working) are presented in Annex 1 and reported in Tab. 4-1: under these conditions, the maximum cryocoolers temperatures in the worst hot scenarios are +30°C, with 10°C positive margin with respect to the requirements.

5.1.3 HEATER FAILURE (ON)

The failure of an heater can be considered as a simple additional power injection on the cryocooler body, which accidentally does not experience the contribution to the temperature drop between cryocooler and evaporator interface.

Therefore it is conservative to consider the plot in Fig. 5-1, in order to calculate the cryocooler and LHP temperatures; in this way, one overestimates the temperature drops considering even the latest contribution through the cryocooler interface.

Since the startup heaters amount is not frozen yet (a test is needed to determine it, see also [RD5]), the worst case assumption of 10W startup heater power can be adopted.

A failure in the worst hot case would lead the following temperature results:

# of LHP working	Cryo Power (W)	Heater fails on:	Cryocooler temperature ²	LHP temperatures (°C – °C) ³	Notes
1	105	Working LHP	<31.6	<<31.6 -- <31.6	1 failure + heater failure
1	105	Not working LHP	31.6	<<31.6 -- 33.4	1 failure + heater failure
2	105	1 Working LHP	<14.1	<<14.1 -- <<14.1	1 Heater failure only
2	105	Both LHP	<18.1	<<18.1 -- <<18.1	2 heaters failure
2	158	1 Working LHP	<34.6	<<34.6 -- <<34.6	1 cryo fails + heater failure

Tab. 5-1: heater failures in different worst hot scenarios (10W startup heater considered).

In the table above, the Cryocooler temperature is always lower than the provided value because the temperature drop evaporator/cryocooler should not be taken into account. The only exception is the case when the heaters fails on a not-working LHP; in that case, the heater power has to cross the two cryocooler interfaces before reaching the working LHP and eventually the radiator.


It is worth noticing that the system, in case 158W are dissipated, and all 4 heaters fail (both lines A and B on both LHP), the peak temperature is 31.3°C at the hottest LHP location (45.6°C on the cryocooler). In the case a single LHP fails with both heaters lines failing on it, the max temperature is 37.2°C on the cryo and 14.5 on the other (working) LHP.

The LHP temperatures (evaporator zone) shall be always lower than the cryocooler temperature whenever the LHP is working. If the LHP is not working, the LHP temperature in the evaporator region is close to the cryocooler one. When the LHP is not working and the heater fails on that LHP, then the evaporator body is 1.8°C hotter than the cryocooler (being the interface conductance equal to 5.5W/K at each interface).

In any case, even in the worst case (158W cryocooler power, 2LHP working), the LHP temperatures and the cryocooler temperatures are always well below 34.6°C, which does not represent any risk neither for the cryocooler, nor for the LHP system.

² The '<' operator is due to the fact that the heater power is considered as applied on the cryocooler body, thus adding a temperature drop contribution at the interface.

³ The '<<' operator is due to the presence of the large temperature drop between the cryocooler temperature of the preceding column, and the LHP evaporator, due to the 5.5W/K interface conductance per evaporator, through which 105÷158W are passing → at least 10°C less

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5.2 COOLING DOWN DUE TO LOSS OF POWER

A second kind of off-nominal cases is represented by the cooling down of the entire experiment due to power outages. The duration of these power outages are considered to last no more than 8 hours.

Due to the presence of the bypass valve, the evaporator and the cryocooler area are decoupled from the radiator zone. Therefore, separate analysis can be carried on these two devices.

5.2.1 CRYOCOOLER COOLING DOWN

In the worst case analysis of the cryocooler, the following assumptions are made:

1. the starting point of the transient 8-hours analysis is the worst cold condition on the ISS, at minimum power level.
2. despite the valve regulation in the cold cases induces a stable cryocooler temperature of about -3°C (see annex 1), the initial temperature of this transient analysis shall be -20°C (valve completely OFF setpoint temperature) + the cryo to evaporator temperature drop ($63\text{W} / (5.5\text{W/K} * 2) = -20 + 5.7 = -14.3^\circ\text{C}$).
3. The heat leakage through the LHP lines material is considered negligible (the transport lines are several meter long, the cross section is few mm^2 and the lines are wrapped by MLI).
4. The Vacuum Case temperature is cooling down as well, and its temperature is calculated by system level model
5. The Inner side of the vacuum case thermal shield is kept at a constant temperature of 70K.
6. The MLI is considered having an effective emissivity of 0.05.

The conductances of the cryocooler to the Vacuum Case and to the cold finger tip (in contact with the 70K thermal shield) are provided by NASA/GSFC. The thermal capacitance is a NASA input as well. Numerical values are recalled here:

GL to cold tip = 0.007 W/K max

GL to Vacuum Case = 0.05 W/K

Cryocooler capacitance = 2124 J/K

It is worth noticing that both conductances to cold tip and to vacuum case are very low when compared to the $2 \times 5.5\text{W/K}$ interface to the LHP. This confirms the correctness of the assumption of disregarding there links in the LHP 2-phase simulations.

Under the assumptions listed above, the temperature profile of the Cryocooler versus time is, in the worst cold case:



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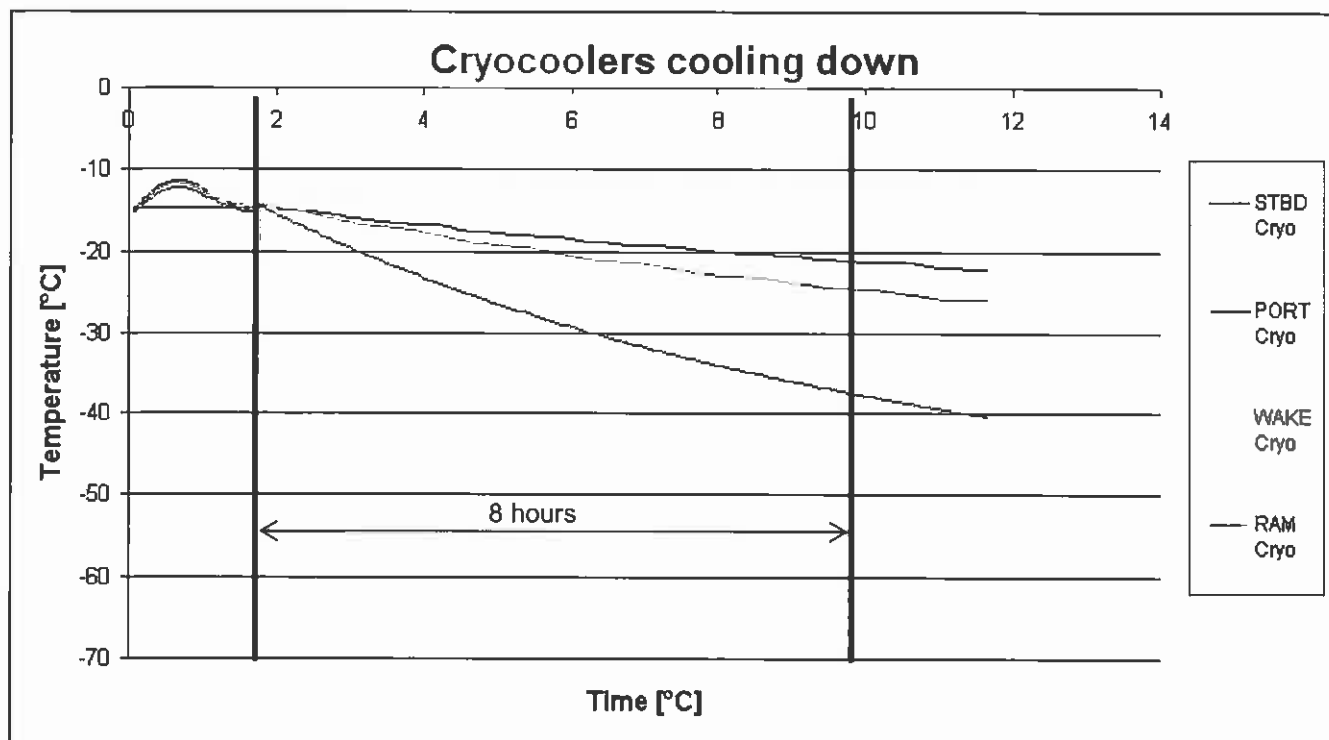


Fig. 5-2: Cryocoolers temperature profile during Cooling down

The cryocoolers temperatures are always above the -40°C limit during the 8 hours of cooling down

5.2.2 ZENITH RADIATOR COOLING DOWN

The cooling down of the Zenith radiator, when no power is applied on it, is reported in the following figure. The simulation results are taken from the system level model, which implements both the spokes and the MLI in between the TRD and the Zenith radiator.

The minimum temperature reached by the outer skin is reported in the following figure; local minimum is -126°C.

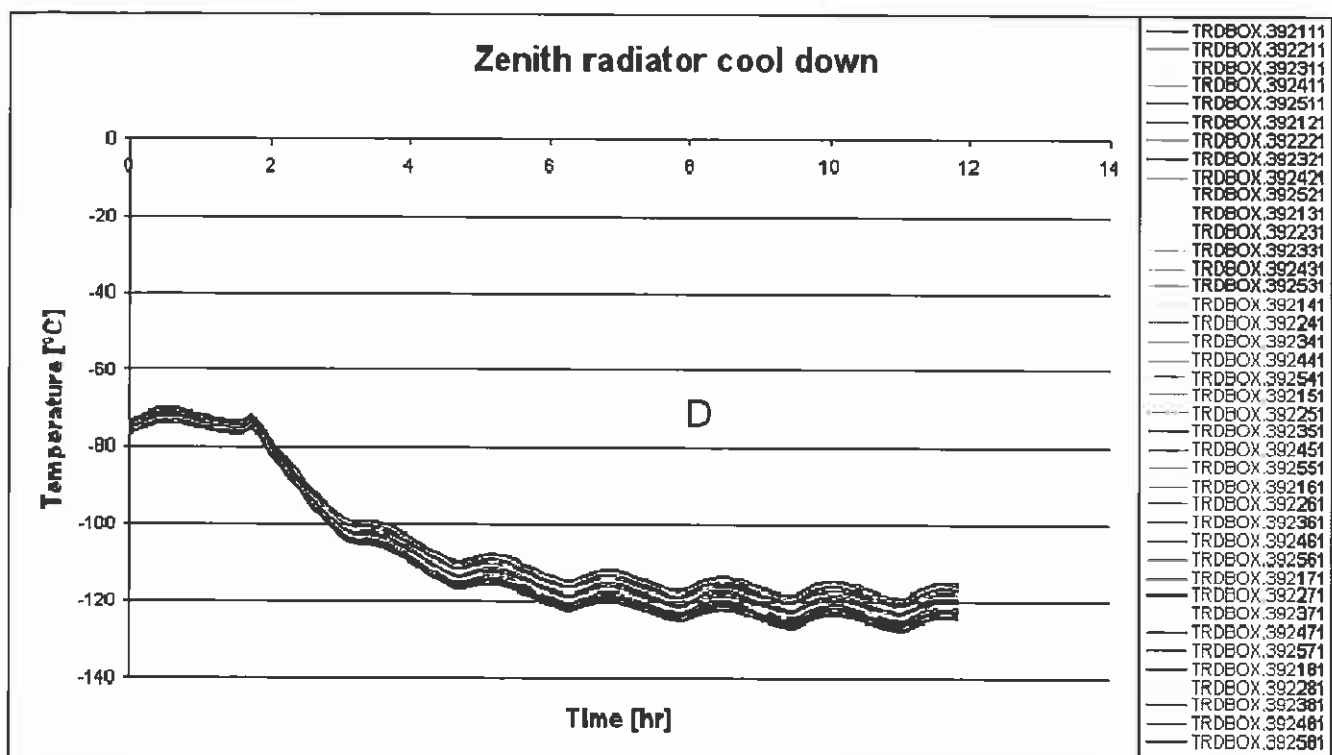


Fig. 5-3: Cooling down Zenith radiator temperature profile

6. SUBSYSTEMS ANALYSIS


6.1 MLI

MLI is covering the cryocooler assembly, LHP included, and shall wrap the transport lines up to the radiator. Due to the low equivalent emissivity, in operational modes the heat leakage through the MLI (in any direction) shall be much lower than the applied heat on the cryocooler.

In fact, considering a 0.16m^2 total MLI surface, external beta cloth (alpha 0.47 and epsilon 0.86 in EOL conditions) the maximum total heat absorbed by the beta cloth layer is $\sim 0.5 \cdot 0.16 \cdot 1422 \cdot 0.47 = 53\text{W}$.

Even assuming that the MERAT of the beta cloth outer surface is equal to the cryocooler internal temperature (strongly conservative assumption, since the beta-cloth has view factor to space), the heat flux irradiated to space and transferred to the cryocooler shall be proportional to the relative emissivities. Being the equivalent emissivity of the MLI equal to 0.05, and the beta cloth one 0.86 in the worst (lowest) case, this means that less than 3 W shall reach the cryocooler through the MLI. This value is 20 times lower than the minimum cryocooler power dissipation, and can be considered negligible for analysis purposes, also considering the high margins attained.

MLI in between the TRD and the Zenith radiator is 1.2m^2 blanket per each radiator quadrant (4.8m^2 total). In general the TRD is always hotter than the zenith radiator.

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- In the cold cases, a maximum gradient of 40°C has been found between the TRD and the radiator (the former at -20°C, the latter at -60°C). This corresponds to a neat heat flux through the MLI equal to 7W. This value has the effect of slightly increasing the radiator temperature, which however shall remain always well below the valve actuation temperature; therefore, no effect is expected in the cold cases onto the cryocooler temperature
- In the hot cases, the TRD shall reach a max temperature around 33°C on its upper cover. Being the average temperature of the radiator equal to -6°C and +7°C in the 105 and 158W heat loads cases respectively, the heat transferred by the MLI from the TRD to the radiator is 12.5 and 8.9W respectively (system level analysis results). This corresponds to a temperature increment at radiator skin level equal to 3.2 and 2.0°C respectively. The broad margins at cryocooler temperature make these corrections negligible.

This explains why in the 2-phase model the MLI is not considered.

6.2 SPOKES DESIGN

The physical interface between the TRD and the zenith radiator is mediated by isolative glass fiber spokes. These spokes are 14 in total per each radiator quadrant, and provide a thermal calculated on the basis of the table below:

E-Glass conductivity	1.3	W/m/K
Epoxy (general)	0.6	W/m/K
Mixture:	60%	fibers
	40%	resin
Ktot:	1.02	W/m/K
Cross Section (3mm diameter)	7.07E-06	m^2
H (length)	6.00E-02	m
Conductance rod	1.20E-04	W/K
Conductance cap (titanium)	1.26E-02	W/K
Conductance (Spoke):	1.19E-04	W/K


Tab. 6-1: spoke conductance evaluation parameters.

The total cumulative conductance of 14 spokes is 1.7E-3 W/K.

Even assuming 100°C gradients between the TRD and the radiator, the transferred heat is less than 1W (0.17W).

This makes the impact on the spokes on the thermal behaviour of the zenith radiator completely negligible.

The spokes thermal design is robust enough to accommodate changes in section and length.

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ANNEX 1. CRYOCOOLER TCS 2-PHASE THERMAL ANALYSIS



IberEspacio
Tecnología Aeroespacial

PROJECT: **CG022**
CLIENT: **CARLO GAVAZZI SPACE**

TITLE:
CRYO COOLER THERMAL MODEL

DOCUMENT No: CG022-NT-0001	ISSUE: Issue 2
ISSUED FOR: INFORMATION	DATE: 14/12/07

Prepared by : ALS / JOK

Reviewed by : RPV / MGX P.A.

Approved by : JOK

QUALITY RECORD

Retention Period: 5 years

MODIFICATION RECORD

Issue	Modification and Reasons for the Modification
1	Initial Issue
2	<p>Page 19: insert pressure and temperature values for Popen and Pclose</p> <p>Page 25: insert "If the simulations had to be repeated with a SINGLE LHP working, the GL should be changed as well, not only the HFM_n"</p> <p>Page 29-31: update with the explanation that the behaviour is due to numerical imprecision, related to the valve modelling and to the small flow rate. The valve is massless in the model, and its answer to pressure changes is immediate (no inertia, no spring). This leads to quick oscillations that are expected to disappear in the real behaviour. The real amplitude and frequency of these oscillations (if any) cannot be predicted until test.</p> <p>Page 28: update the table with the capacitance values</p>

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1. ABSTRACT

The Thermal model described in this document has been updated taking into account the last design changes of the LHP's to be mounted in AMS-02 that update the previous models specified in the Reference Documents.

The thermal Interface parameters provided by Carlo Gavazzi Space, such as sink Temperatures, orbital heat loads and radiation exchange factors have been considered as the input of this thermal model.

The obtained Results of the model are included in this document with the final conclusions issued from this results.

2. DOCUMENTS

2.1 APPLICABLE DOCUMENTS

[AD01] AMS02-SP-0008_iss1 AMS-02 CRYO Loop Heat Pipe Specification

2.2 REFERENCE DOCUMENTS

[RD01] CG005-NT-0001 Issue Draft Thermo-Hydraulic Analysis of the AMS Cryocooler LHP

[RD02] <http://www.ecosimpro.com/>

[RD03]: NIST (National Institute of Standards and Technology), March-1990.

[RD04] Interface Data from Carlo Gavazzi Space email received on friday 14/09/2007

[RD05] Condenser Length data email received on 10/10/2007.

3. ACRONYMS, ABBREVIATIONS AND FORMULATION NOMENCLATURE

3.1 ACRONYMS AND ABBREVIATIONS

CGS:	Carlo Gavazzi Space
DN:	Diffusion Node
HEM:	Homogeneous Equilibrium Model
IE:	IberEspacio
LHP:	Loop Heat Pipe
PVTCD:	Passive Variable Thermal Conductance Device

3.2 FORMULATION NOMENCLATURE

dP_{loss} :	Pressure losses in the wick
k_{eff} :	Effective thermal conductivity of the wick.
L_w :	Length of the wick.
\dot{m}_{evap} :	Evaporated mass flow.
\dot{m}_{CC} :	Mass flow exchanged between the evaporator and the compensation chamber.
\dot{m}_{LL} :	Mass flow exchanged between the evaporator and the liquid line.
M_f :	Working fluid mass.
\dot{Q}_{cc} :	Total heat transfer from the evaporator to the compensation chamber.
\dot{Q}_2 :	Heat leak to the compensation chamber.
\dot{Q}_{sc} :	Subcooling heat.
V_{AG} :	Axial grooves volume.
V_{CG} :	Circumferential grooves volume.
V_{CC} :	Compensation chamber volume.
V_{cond} :	Condenser volume.
V_{LHP} :	Loop heat pipe volume.
V_{VL} :	Vapour line volume.
κ :	Permeability of the wick.
λ :	Latent heat of vaporization.
μ_l :	Viscosity of the saturated liquid.
ρ_l :	Density of the saturated liquid.
ρ_g :	Density of the saturated gas.

4. INTRODUCTION

4.1 AMS CRYOCOOLER THERMAL DESIGN

AMS includes 4 Stirling Cryogenic Coolers (Cryo-Coolers), which extract parasitic heat from one of the thermal production shields, which are located around the the Helium cooled AMS magnet (see Figure 4-1).

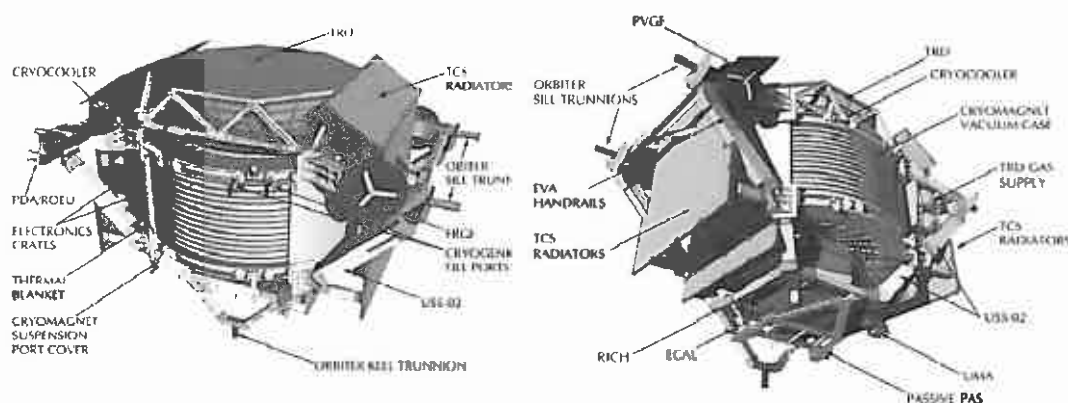


Figure 4-1 : AMS

LHPs are employed to transfer the dissipative heat from the cryo-coolers to the dedicated thermal radiator (see Figure 4-2)

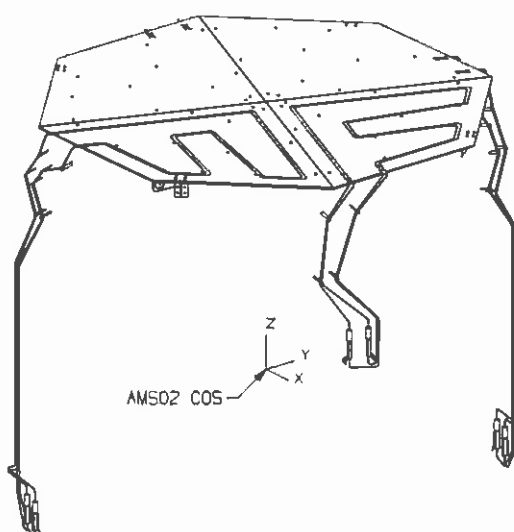


Figure 4-2 : AMS Cryo-cooler evaporators and radiator

The thermal design of each cryo-cooler includes the following items:

- Two LHP evaporators with integrated reservoir and by-pass valve.
- Transport lines (liquid & vapor).
- Condenser connected to the Zenith radiator panel.
- Heater, thermostats, temperature sensors attached to evaporator block.
- Temperature sensors on radiator panels.

The working fluid is propylene, which has a freezing temperature of -189°C and avoids the complexity of designing a freeze tolerant radiator.

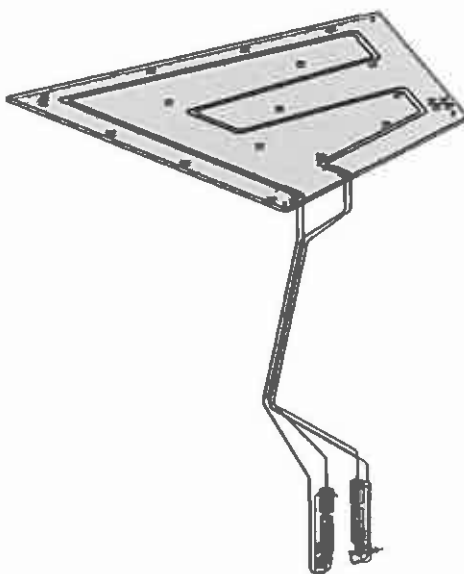


Figure 4-3 : *Two LHP circuits and Radiator panel of an AMS Cryocooler*



Figure 4-4 : *Evaporator block of the Cryo-cooler LHPs*

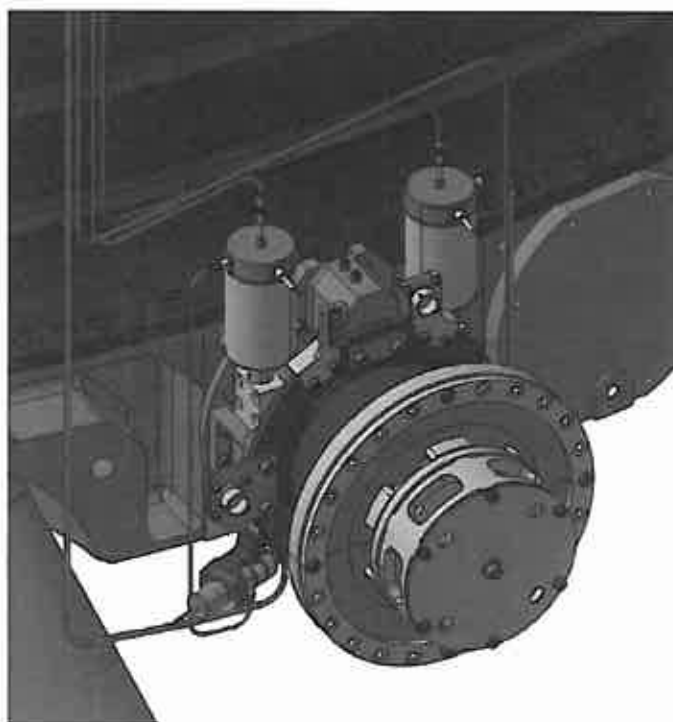
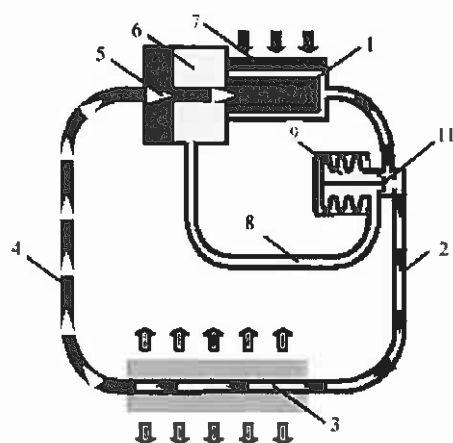


Figure 4-5 : *AMS Cryocooler with two evaporators mounted on it*

The design includes a passive by-pass valve to protect the cryo-cooler from low temperatures.



1 - porous wick; 2 - vapour line; 3 - condenser; 4 - liquid line.
5 - part of liquid line inside the reservoir; 6 - reservoir; 7 - LCT.
8 - bypass line; 9 - bellows; 10 - heater; 11 - valve.

Figure 4-6 : *Lay out of the LHP with a passive controlled valve*

The LHP systems have been divided, according to their localization on AMS in WAKE and RAM LHP's system, Where

- RAM defines the objects at the $-Y$ hemi-space of AMS
- WAKE defines the objects at the $+Y$ hemi-space of AMS.

Taking into account the position of the LHP, the LHP's systems have been divided in upper and lower LHP's as showed in the Figure 4-7:

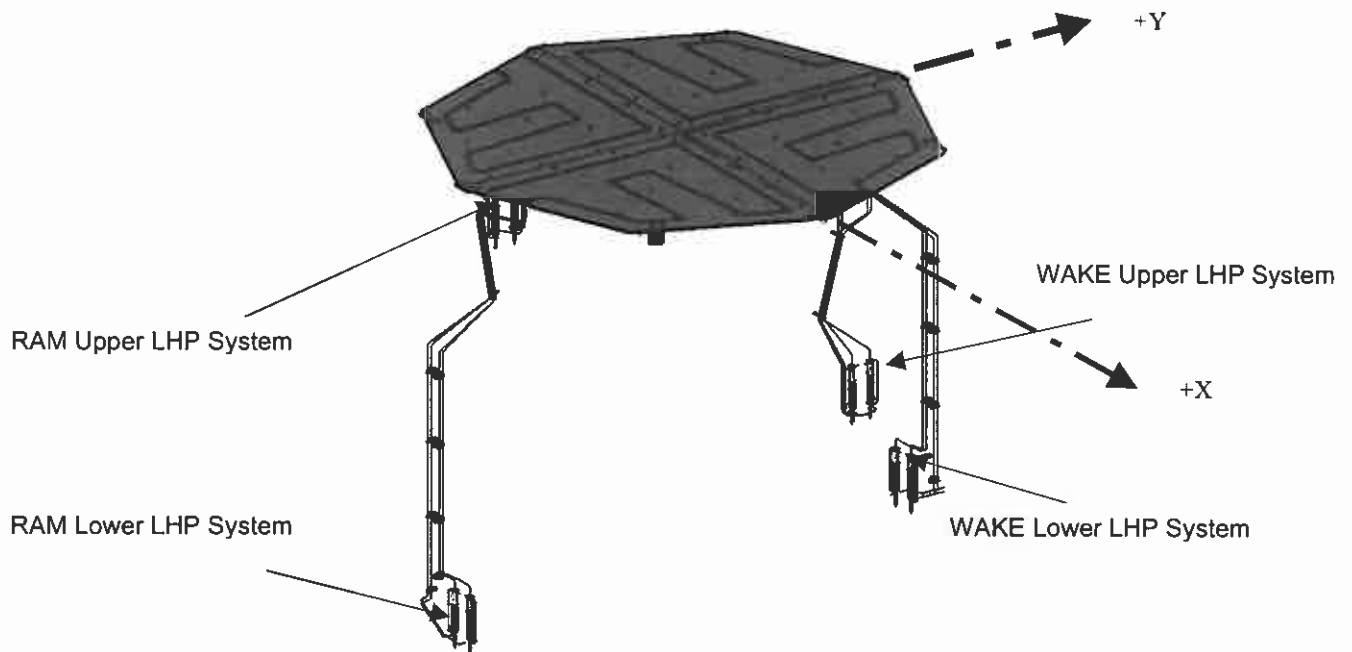


Figure 4-7 : *UPPER and LOWER LHP's Location*

The main objective of this thermal analysis is to consolidate the design of these LHP's.

4.2 AMS CARLO GAVAZZI SPACE CRYOCOOLER MODEL INPUT DATA

The LHP has been designed taking into account the next parameters and inputs provided by Carlo Gavazzi Space.

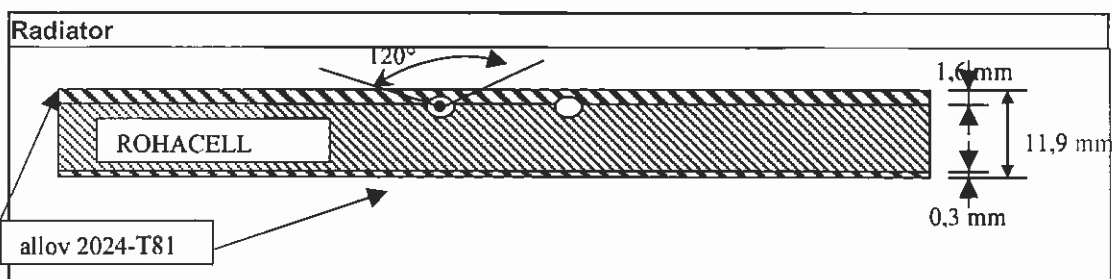
Capillary Pump			
	Description	Value	Units
Wick properties	Wick material	Nickel	-
	Wick external diameter	0,014	m
	Wick length	0,1	m
Axial Grooves	Number of axial grooves	4	-
Evaporator	Evaporator heated length	0,1	m
Saddle Material	Saddle dimensions (length/width/ height)	100*30*0,05	mm ³
	Contact area x contact conductance saddle-cryocooler	5,5	W/K
Thermal Environment	Sink equivalent temperature for this area	Isolated	K
	GRs between sink and capillary pump	Isolated	m ²
	Load + Source heat flows (excluded Elec. Box heat load)	Isolated	W
Working Fluid	Propylene		

Vapor Line	
Thermal Environment	ISOLATED

Liquid Line	
Thermal Environment	ISOLATED

Condenser Line			
Size	Condenser Line Wall Thickness	0,0005	m
	Condenser Line Length	4,962	m
	Condenser Line Hydraulic Diameter	0,003	m
Bending	Total number of bends	8	-
	Angle of bend number 1	90	°
	Bending radius of bend number 1	0.04	m
	Angle of bend number 2	67,5	°
	Bending radius of bend number 2	0.04	m
	Angle of bend number 3	119,1	°
	Bending radius of bend number 3	0.04	m
	Angle of bend number 4	96,6	°
	Bending radius of bend number 4	0.04	m
	Angle of bend number 5	86,4	°
	Bending radius of bend number 5	0.04	m
	Angle of bend number 6	64	°
	Bending radius of bend number 6	0.04	m
	Angle of bend number 7	135	°
	Bending radius of bend number 7	0.04	m
	Angle of bend number 8	112,5	°
	Bending radius of bend number 8	0.04	m

Additional Losses	Local Hydraulic Resistance Coefficients (excluding bends and flat capillary isolator)	7	-
Radiator Interface	Conductance per unit length condenser tube-radiator plate	380	W/m K
Thermal Environment	Sink equivalent temperature for radiator nodes	See I/F Data	K
	GRs between sink and radiator nodes	See I/F Data	m ²
	Load + Source heat flows	See I/F Data	W



Valve		
Valve Set Point	-20	°C

Table 4-1 : *Input Data*

4.2.1 Radiator inputs

The I/F data for the radiator, i.e., GR, sink temperature, incident heat fluxes...have been provided for the radiator shown in Figure 4-8.

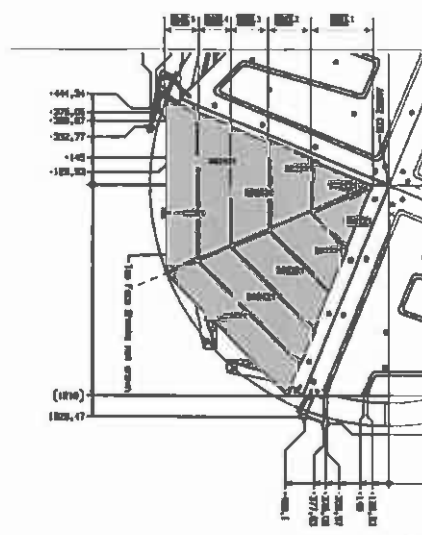


Figure 4-8 : Radiator discretization provided by CGS

The summary of the input data related to this radiator are:

	Description	Value for analysis
Condenser	Line wall thickness	0.0005m
	Condenser Line length	4.962m
	Condenser line hydraulic diameter	3mm
	Conductance per unit length between condenser tube and radiator plate	380W/mK

Table 4-2 : Condenser Analysis Input Values

Each radiator hosts two LHP's, as described in previous sections, an "internal" one and an "external" line having different lengths.

Taking into account that these lengths differences are no critical in the model to predict the performances, an unique condenser line length has been considered for thermal modelisation purposes and it has been stated the same condenser line length for the eight different LHP's. This length correspond to the average value of the "internal" and "external" line length (see Table 4-3)



	Length brazed to radiator (mm)	floating lines outside the radiator (up to joint included) (mm)	TOTAL (mm)
External Line	4813	90+90	4993
Internal Line	4730	90+90	4910
Average			4962

Table 4-3 : *Condenser Length Input Values*

The values of the sink equivalent temperatures, the radiative thermal couplings and the incident heat fluxes for each radiator node have been provided as input data for each orbital position . All these parameters are included in the associated paragraphs for the Hot and Cold Case.

4.2.2 Valve Set Point

The set point of the valve is defined as the temperature at which the condenser line is completely closed, and the entire flow (in nominal conditions) is directed back to the Compensation Chamber through the bypass line.

4.3 THERMAL REQUIREMENTS

The Thermal Requirements from [AD01] are:

- AMS-CRYO-LHP-3.1.2 -010

Heat Transport Capability : *"The LHP system shall carry cryo cooler own dissipation (from 60 W to 150W)+ the heat lift (from 3 W to 8 W)."*

- AMS-CRYO-LHP-3.1.2 -020

Nominal Operations : *"For design purposes each LHP shall have a Heat Transport Capability (HTC) of 100 W +5 W heat lift =105 W with the evaporator in the temperature range -30°C to +30°C"*

- AMS-CRYO-LHP-3.1.2 -030

Contingency Operations : *"To take into account either the failure of a LHP or a LHP not starting each LHP shall be capable of carrying a maximum power of 137 W at a maximum temperature of 30 °C (on the evaporator)."*

Results of the calculations of the maximum heat fluxes of the LHP's provide a value of around 150W. Taking into account that it is a calculation having an error margin , the external maximum power ability of this LHP could be slightly less than the required value of 137W. The real value of this maximum heat transport ability could be measured by test.

4.4 THERMAL DESIGN CASES

4.4.1 Hot Case

Power = 158 W

The HOT CASE input data are included in Annexe I.

4.4.2 Cold Case

Power = 63 W

The COLD CASE input data are included in Annexe II.

5. THERMAL MODEL

5.1 ECOSIMPRO LIBRARY ASSUMPTIONS

To design the LHP, a model has been developed using the mathematical tool EcosimPro [RD02]. This model has been also used to verify LHP performances. To fully understand the model, an overview of the mathematical formulation used for the modelling is given hereafter.

5.1.1 Mathematical formulation

The mathematical model has been built modularly. This means that the complete LHP model has been created connecting the typical LHP components: evaporator, compensation chamber, transport lines, condenser and pressure regulator valve. The main assumptions made in all the components are as follows:

- The 1D fluid flow model is a homogeneous equilibrium model (HEM). That is, the one-dimensional conservation equations are established for the two-phase mixture and the amount of vapour and liquid is taken into account in terms of quality. It is considered that the two phases are in equilibrium assuming equal phase velocities, temperatures and pressures.
- The calculated thermodynamic properties correspond to the two-phase mixture. However, in saturation conditions the thermodynamic properties are calculated separately for vapour and liquid phases. The thermodynamic properties are obtained by interpolation using the tables built from NIST [RD3] routines.
- The LHP components can exchange heat with the environment by convection and radiation, and they can exchange heat with other external components (such as saddles) by conduction.
- The fluid model is based on the one dimensional fundamental conservation equations (mass, momentum and energy) applied to control volumes. The fluid part of each LHP component is sub-divided into individual control volumes.
- The compressibility and transient effects are also taken into account. The viscous effects are taken into account through the pressure drop calculations. The pressure drop calculations use the built-in EcosimPro functions for different elements. These include empirical correlations for the pressure drop in a porous media.
- The gravity effects due to the different orientations of the LHP have been taken into account in the formulation.

PRESSURE REGULATOR VALVE

The mass and energy conservation equations are applied for the fluid contained in the valve in order to obtain the density and the internal energy at any time. To fix the valve set point, the model requires an input temperature value (T_{open}). The saturation pressure corresponding to this temperature (P_{open}) is obtained by interpolation using the tables from NIST and represents the point

where the valve open completely the bypass path and, consequently, the radiator branch is completely closed. Taking into account the properties of the valve bellows, a new pressure (P_{close}) is calculated to define the point where the valve starts to open the bypass path and part of the mass flow goes through the radiator and part of the mass flows through the bypass line to the Compensation Chamber.

The valve position is calculated depending on the relation between the fluid pressure and these values of P_{open} and P_{close} as follows:

$$\begin{aligned} pos &= 0 & P_{fluid} &\geq P_{close} \\ 0 < pos < 1 & & P_{close} > P_{fluid} > P_{open} \\ pos &= 1 & P_{fluid} &\leq P_{open} \end{aligned}$$

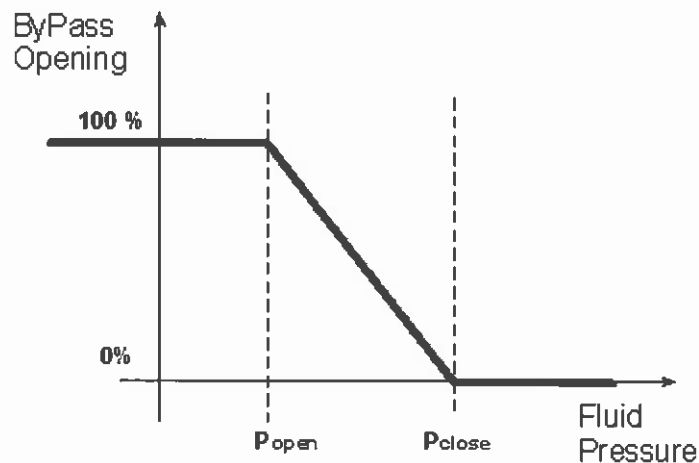


Figure 5-1 : Valve position

The values for P_{open} and P_{close} with their corresponding temperatures are in this case:

- **$P_{open} = 3.09\text{bar} \rightarrow -20.0^\circ\text{C}$**
- **$P_{close} = 4.5\text{bar} \rightarrow -8.5^\circ\text{C}$**

Therefore, in regulation conditions, it is assumed that the valve reaches a steady state in an intermediate position between 0 and 1.

Finally, the momentum equation is solved for each of the branches (direct and bypass) taking into account the value of the valve position.

5.2 LHP THERMAL ANALYSIS

5.2.1 LHP model description

5.2.1.1 Geometry of the Equivalent Radiator

The design of the radiator is such that there may be significant conduction from the two-phase zone of the condenser tube to the sub cooled portion of the condenser tube.

The nodalization of the radiator model provided by CGS may be too large to account properly the conduction between the two-phase portion and the subcooled portion of the condenser line.

A detailed radiator model has been built to make a good estimation of the conduction effect (see [RD01]), this detailed model is made of rectangular plate components connected together. The actual radiator is not rectangular, however we have defined an equivalent rectangular radiator based on our engineering judgment that should have very approximately the same behaviour than the actual radiator.

Equivalent Radiator Layout

The geometries of the actual radiator and the equivalent rectangular radiator are shown in Figure 5-2.

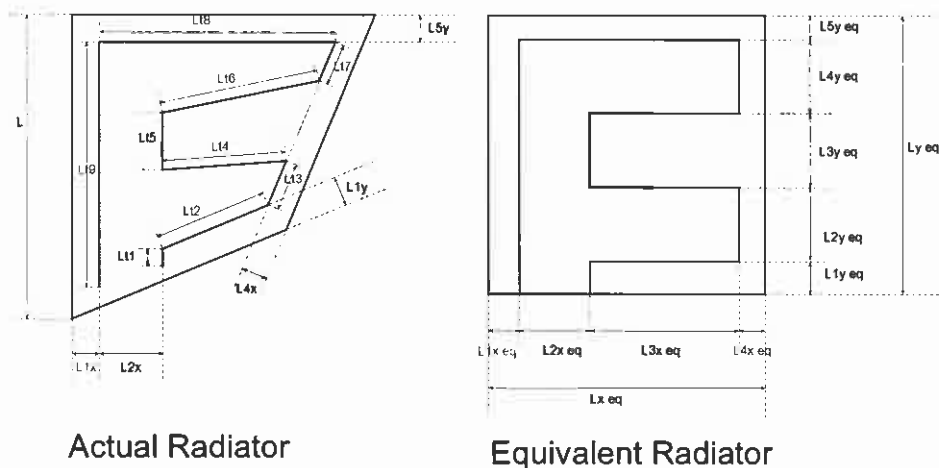


Figure 5-2 : *Original and Equivalent Radiator*

The description of the assumptions and calculation of this equivalent radiator model have been widely commented in [RD01].

The only difference with the model described in [RD01] is that the conductance per unit length between condenser tube and radiator plate is settled to 380W/mK (by CGS input) and thus, several GL's have been added in the new ecosim radiator model and their conductance have been calculated according to this input data.

5.2.1.2 EcosimPro Radiator Model

The radiator model is basically made of rectangular plates where two-D conduction and radiation losses are accounted and fluid tubes connected to the radiator plates via HF multipliers and GL components. A simplified sketch of the EcosimPro model of the radiator is shown in *Figure 16*, where the discretization that have been used for the different radiator plates and tubes is also shown (see dashed lines). It must be noticed that the multiple thermal connections on each side of the radiator plate can be made through an unique thermal port, because EcosimPro has a vectorized thermal port that can exchange an array of temperatures and array of heat flows to represent the thermal contact along a line.

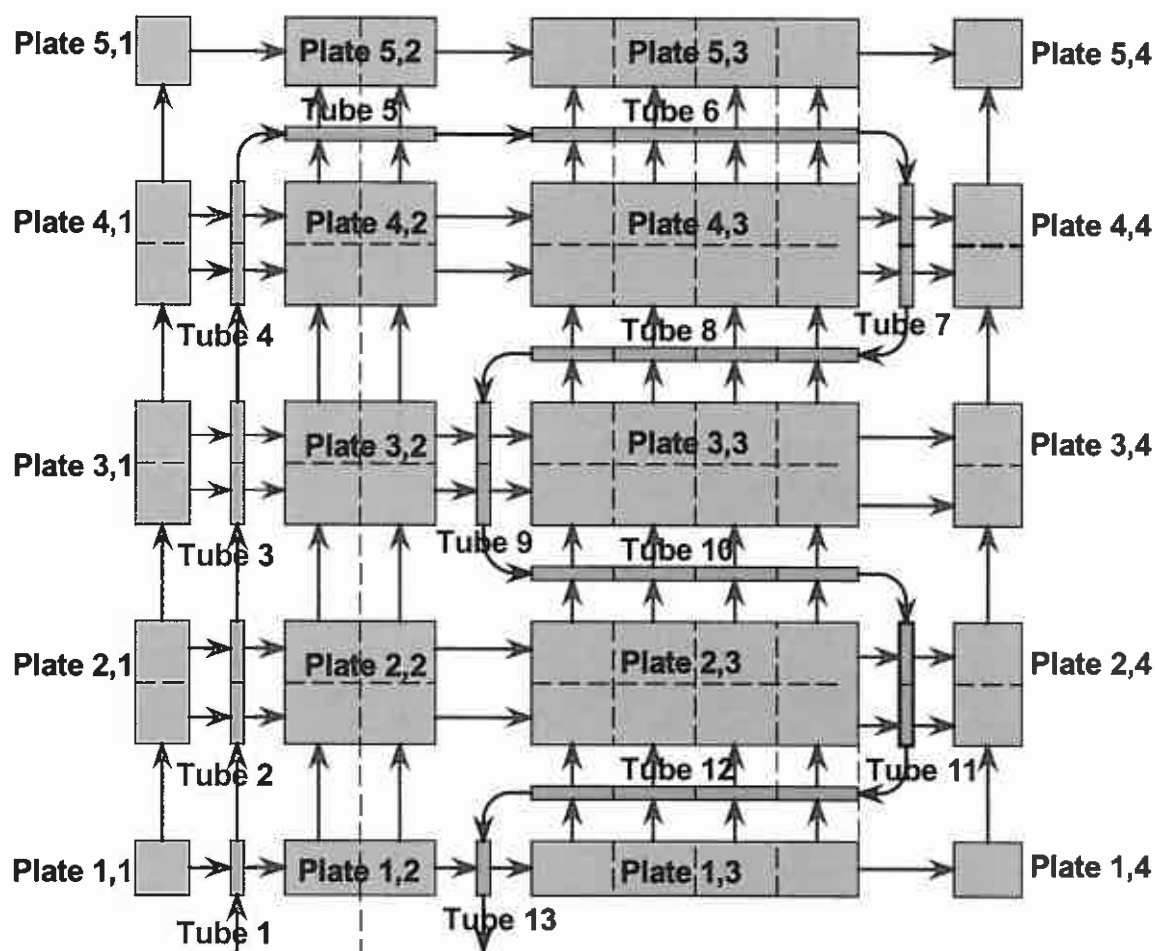


Figure 5-3 : Sketch of Detailed Equivalent Radiator Model

The actual model only represents one Loop Heat Pipe by taking into account that both LHP's should behave approximately the same way because the layout of the two LHP circuits is approximately the same as discussed in previous paragraphs.



As only one LHP is represented in the model, the heat flow extracted by one LHP from the electronic unit has to be divided by two to account that the other LHP is not represented. This is done by the component of the type HFM (Heat Flow Multiplier) labelled as HFM_1.

At the radiator side, the heat from the LHP to the radiator plate has also to be doubled to account for the second LHP (see components HFM_1 to HFM_13. in).

The final flow-sheet of the radiator model is slightly more complex because some additional components are needed. For example, the radiator borders have to be isolated and GL components must be introduced between the tubes and the plates to represent the thermal contact between the LHP condensers and the radiator. The actual flow sheet of the detailed radiator model is shown in Figure 5-4.

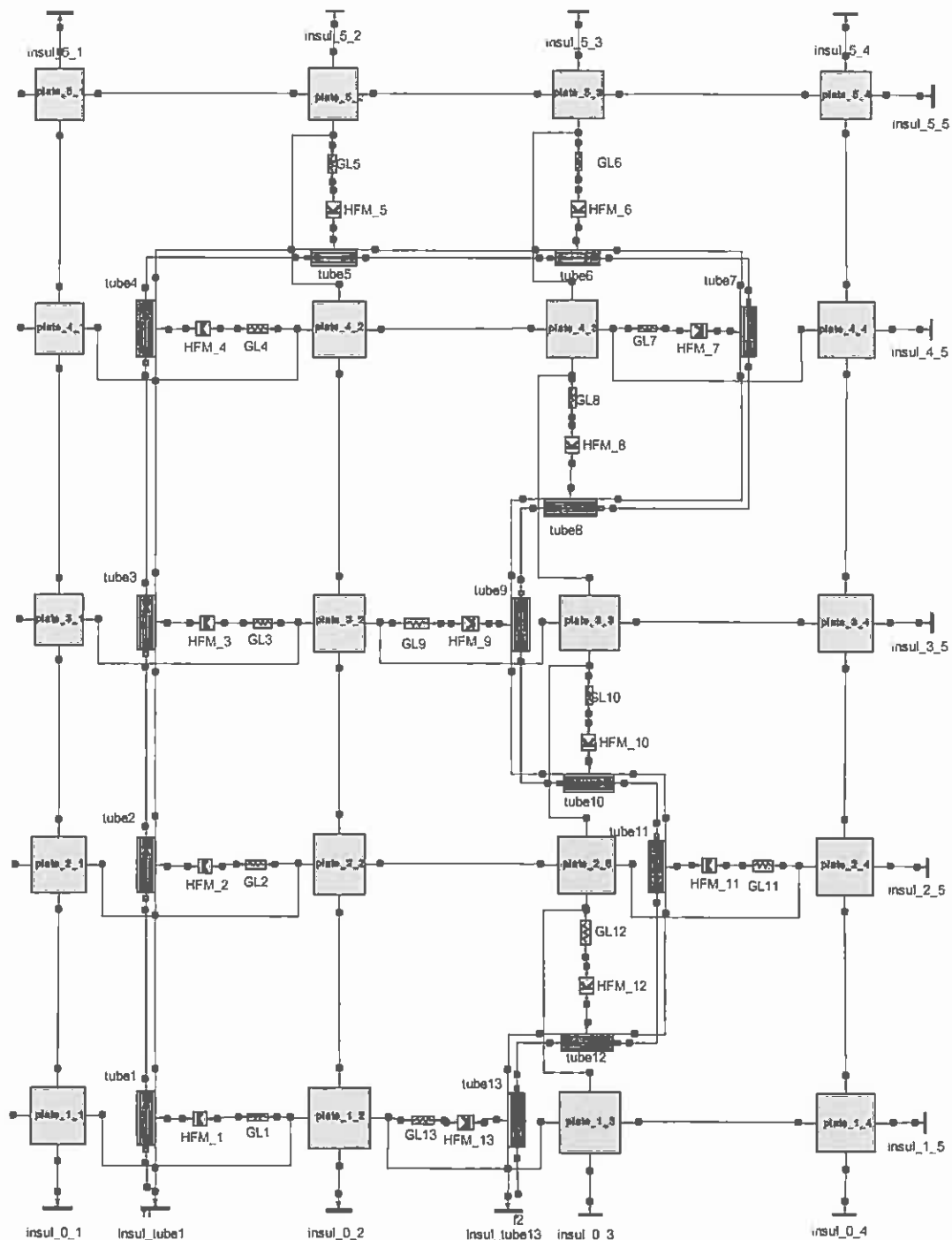


Figure 5-4 : *Flowsheet of the EcosimPro model for the Detailed Equivalent Radiator*

The name of this component is `Zenith_Radiator` and its symbol for including it in other models is shown in Figure 5-5.

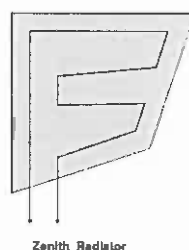


Figure 5-5 : Symbol of the Detailed Equivalent Radiator (*Zenith_Radiator* component)

The LHP detailed radiator model is shown in Figure 5-6

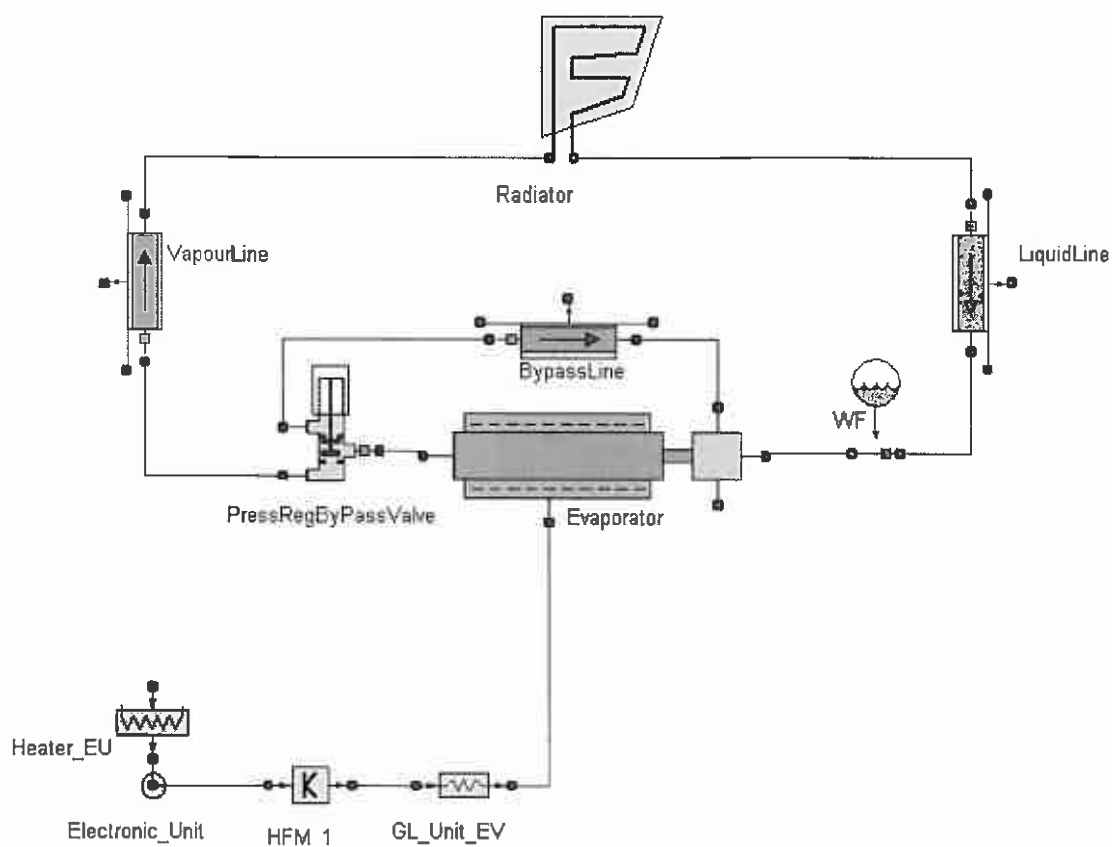


Figure 5-6 : Flowsheet of the EcosimPro model of the AMS02 Cryocooler LHP with Detailed Radiator

5.2.2 LHP model assumptions

To consolidate the design of the Cryo-cooler, an LHP model has been developed using the previous mathematical formulation and assuming the following conditions:

- The LHP components do not exchange heat with the ambient. (input CGS)
- The cryocooler is a single thermal diffusion node.
- The couplings in the condenser/radiator interface have been calculated considering the input data of 380W/mK (input CGS).
- The thermal contact conductance considered in the evaporator saddle/dissipating unit interface is 5,5·W/K according to the input data provided by CGS.
- The thermal coupling between the evaporator and the compensation chamber is set to 0.1 W/K.

All the test correction factors have been set to one.

This model only represents one loop heat pipe by taking into account that both LHP's should behave approximately the same way because the layout of the two LHP circuits is approximately the same.

If only one LHP is represented in the model, the heat flow extracted by the LHP from the electronic unit has to be doubled to account for the non-represented LHP, this is done by the component of the type HFM (Heat Flow Multiplier) labelled as HFM_1.

At the radiator side, the heat from the LHP to the radiator plate has also to be doubled to account for the second LHP (see components HFM_1 to HFM_13. in).

NOTE: *IF the simulations had to be repeated with a SINGLE LHP working, the GL should be changed as well, not only the HFM_n*

In this model it is considered that the equivalent sink temperature in the radiator is the same at each orbital time for all the nodes, and it is calculated such as the average of the nodal sink temperatures provided by CGS. In the same way, the thermal radiative couplings (GR) per unit area are the same for all the plates at each orbital time and corresponds to the sum of the nodal GR provided by CGS. Finally, it is calculated the sum of the nodal incident heat fluxes provided by CGS at each orbital time. The heat value introduced in each radiator node of the model corresponds to the heats sum averaged by the node area.

5.3 UPPER LHP

5.3.1 Input Data for the Model:

Both LHP systems (RAM and WAKE) are identical. As there are not significant differences between the two LHP's of one side, the analysis has been only performed on one of them. For reference, the data introduced in the model corresponds to the Upper RAM LHP System Assy.

Taking into account the assumptions showed in the last paragraph, we include hereafter the input data for the upper LHP model:

- LHP_Detailed Radiador DATA:

DESCRIPTION	VALUE	UNITS
Initial Temperature of the LHP components	280	K
LHP Working fluid	Propylene	-

- COMPONENT – CRYOCOOLER

DESCRIPTION	VALUE	UNITS
Heat capacity	2124	J/K
Initial temperature	283	K

- COMPONENT – EVAPORATOR

DESCRIPTION	VALUE	UNITS
Heat capacity of the compensation chamber case	28	J/K
Elevation of the compensation chamber	0	m
Thermal coupling between the evaporator and the compensation chamber	0.1	W/K
Superheating required for evaporation start-up	0.1	K
Number of axial grooves	4	-
Heat capacity of evaporator case	12	J/K
Evaporator thickness	0.0005	m
Material of the evaporator case	Stainless Steel	-
Number of energy nodes in the wick	6	
Vertical coordinate of evaporator center	0	m
Shape of the axial grooves	Rectangular	-
Wick external diameter	0.014	m

DESCRIPTION	VALUE	UNITS
Primary wick material	Nickel	
Heat capacity of the saddle	128	J/K
Thermal conductance between the saddle and the evaporator case	38.6	W/K

- COMPONENT – GL CRYOCOOLER – EVAPORATOR SADDLE

DESCRIPTION	VALUE	UNITS
Total thermal conductance	5.5	W/K

- COMPONENT – BYPASS LINE

DESCRIPTION	VALUE	UNITS
Inner diameter of the pipe	0.0015	m
External diameter of the pipe	0.002	
Total pipe length	0.314	m
Radius of curvature of each bend	0.01,0.01,0.01	m
Angle of each bend	90,90,90	°
Number of bends	3	-
Nodes of the pipe	2	-
Wall material	Stainless Steel	-
Elevation of the second fluid port	0	m

- COMPONENT – BYPASS VALVE

DESCRIPTION	VALUE	UNITS
Temperature to open completely the bypass branch	253.15	K
Bottom elevation relative to a z fixed axis	0	m

- COMPONENT – VAPOUR LINE

DESCRIPTION	VALUE	UNITS
Inner diameter of the pipe	0.003	m
External diameter of the pipe	0.004	
Total pipe length	1.257	m
Radius of curvature of each bend	0.02,0.02,0.02,0.02,0.02,0.015	m
Angle of each bend	100.13,92.93,108.97,120.72,90,90	°
Number of bends	6	-

DESCRIPTION	VALUE	UNITS
Nodes of the pipe	5	-
Wall material	Stainless Steel	-

- COMPONENT – LIQUID LINE

DESCRIPTION	VALUE	UNITS
Inner diameter of the pipe	0.002	m
External diameter of the pipe	0.003	
Total pipe length	1.163	m
Radius of curvature of each bend	0.015,0.015,0.015,0.015	m
Angle of each bend	94.18,93.5,98.18,110.38	°
Number of bends	4	-
Nodes of the pipe	5	-
Wall material	Stainless Steel	-
Elevation of the second fluid port	0	m

- COMPONENT – HEAT FLOW MULTIPLIER

DESCRIPTION	VALUE	UNITS
Heat flow multiplier, i.e. outlet heat / input heat	0.5	-

- COMPONENT – RADIATOR

DESCRIPTION	VALUE	UNITS
Number of nodes X axis	4	-
Number of nodes Y axis	5	-
Width	0.0016	m
Node Length Lx1 (seeFigure 5-2)	0.116	m
Node Length Lx2 (seeFigure 5-2)	0.223	m
Node Length Lx3 (seeFigure 5-2)	0.694	m
Node Length Lx4 (seeFigure 5-2)	0.107	m
Node Length Ly1 (seeFigure 5-2)	0.129	m
Node Length Ly2 (seeFigure 5-2)	0.273	m
Node Length Ly3 (seeFigure 5-2)	0.273	m
Node Length Ly4 (seeFigure 5-2)	0.273	m
Node Length Ly5 (seeFigure 5-2)	0.107	m
Heat Flow Multipliers (1 to 13)	2	-
Total capacitance of the radiator skin	1346	J/K

5.3.2 Scenario

Although the input requirements only refers to max Power for the Hot Case and min Power for the cold case, the model includes, for both cases the next power input profile: first, the nominal dissipation (105 W) is introduced in the Electronic Unit. After six orbital periods, the power is set to the maximum (158 W). Finally, after six orbital periods the power is set to the minimum dissipation (63 W). The power sequence is then : 105 W, 158 W, 63 W.

5.3.3 Results

5.3.3.1 Hot Case

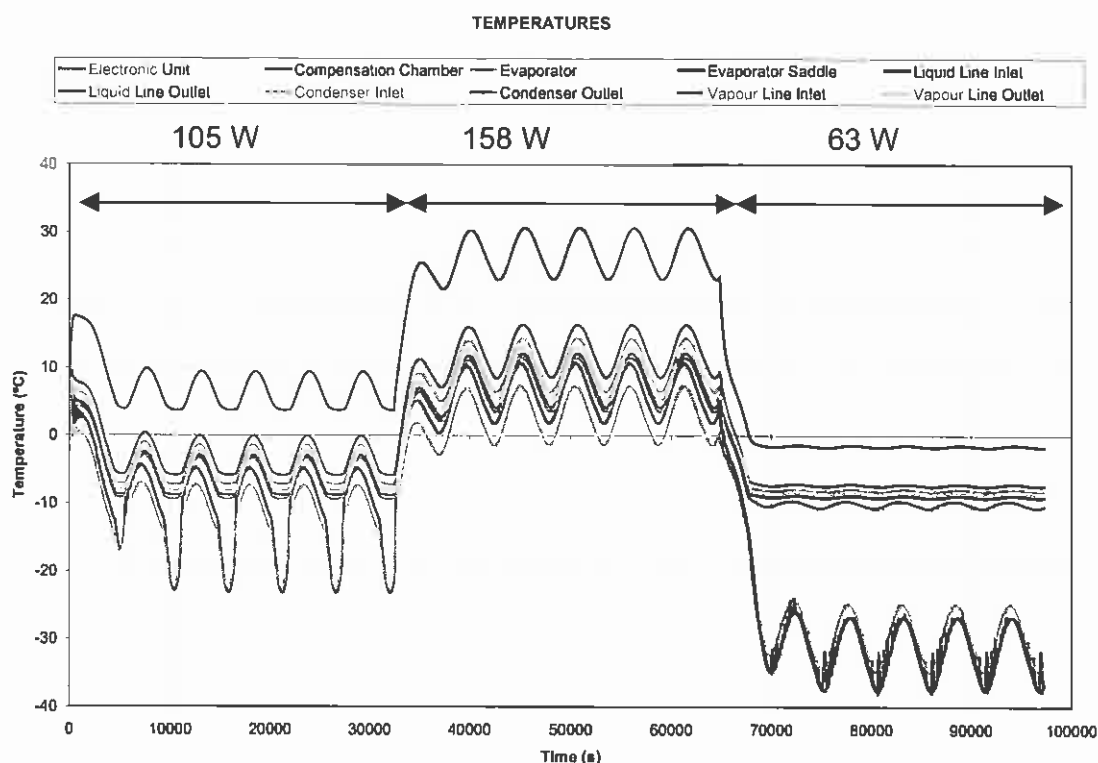


Figure 5-7 : Upper LHP temperatures for the Hot Case.

Figure 5-7 represents the temperatures of the Electronic Unit, Compensation Chamber, Evaporator, Transport Lines and the temperatures at the inlet and outlet of the condenser.

For the three studied cases (applied powers 105, 158 and 63 Watts) the temperature of the evaporator saddle is within the required temperature range (-30°C to +40°C).

Additionally, the valve is regulating the temperature when applying 105 and 63 Watts. When applying 105 W, the valve is alternatively partially opened and closed, producing a "plate in the oscillation of the evaporator saddle temperature. When 63W are applied, the valve is always partially open, that is, part of the mass flow goes through the radiator and the other part is by-passed to the compensation chamber.

In particular, the temperature of the evaporator saddle in this hot case when applying the maximum power (case which is referred in the requirements as indicated previously) oscillates between 16.1 and 8.9 °C without any regulation of the valve.

We can see from the figure that when the by-passed mass flow increases, the decoupling between the "hot side" and the "cold side" are also increased.

The radiator temperatures are represented in the following figure:

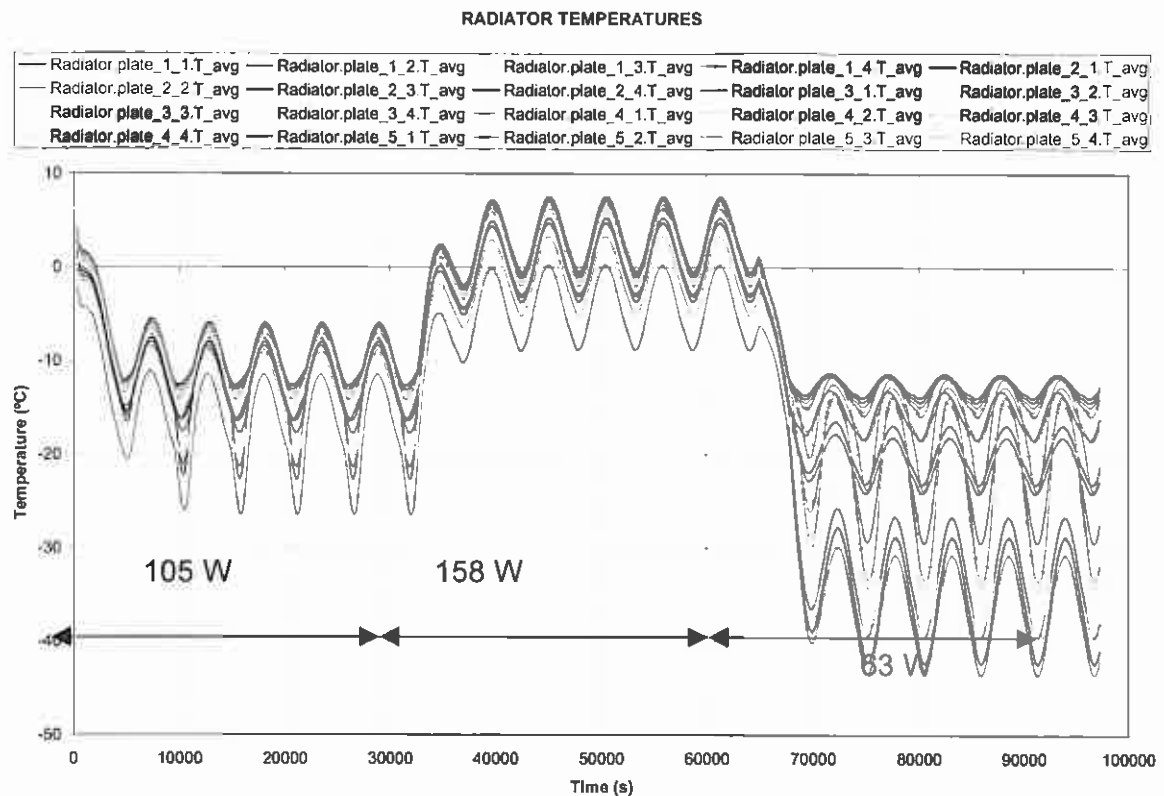


Figure 5-8 : Upper LHP Radiator temperatures for the Hot Case

The maximum and minimum temperatures calculated for hot conditions are listed in the following table:

Power →	105 W	158 W	63 W
Electronic Unit	$T_{\max} = 9.7\text{ }^{\circ}\text{C}$	$T_{\max} = 30.5\text{ }^{\circ}\text{C}$	$T_{\max} = -1.3\text{ }^{\circ}\text{C}$
	$T_{\min} = 3.7\text{ }^{\circ}\text{C}$	$T_{\min} = 23.1\text{ }^{\circ}\text{C}$	$T_{\min} = -1.7\text{ }^{\circ}\text{C}$
Evaporator Saddle	$T_{\max} = 0\text{ }^{\circ}\text{C}$	$T_{\max} = 16.1\text{ }^{\circ}\text{C}$	$T_{\max} = -7.1\text{ }^{\circ}\text{C}$
	$T_{\min} = -5.9\text{ }^{\circ}\text{C}$	$T_{\min} = 8.9\text{ }^{\circ}\text{C}$	$T_{\min} = -7.4\text{ }^{\circ}\text{C}$

Table 5-1 : Upper LHP Electronic Unit and Evaporator Saddle approximate maximum and minimum temperatures for the Hot Case

5.3.3.2 Cold Case

Figure 5-9 represents the temperatures of the Electronic Unit, Compensation Chamber, Evaporator, Transport Lines and the temperatures in the inlet and outlet of the condenser.

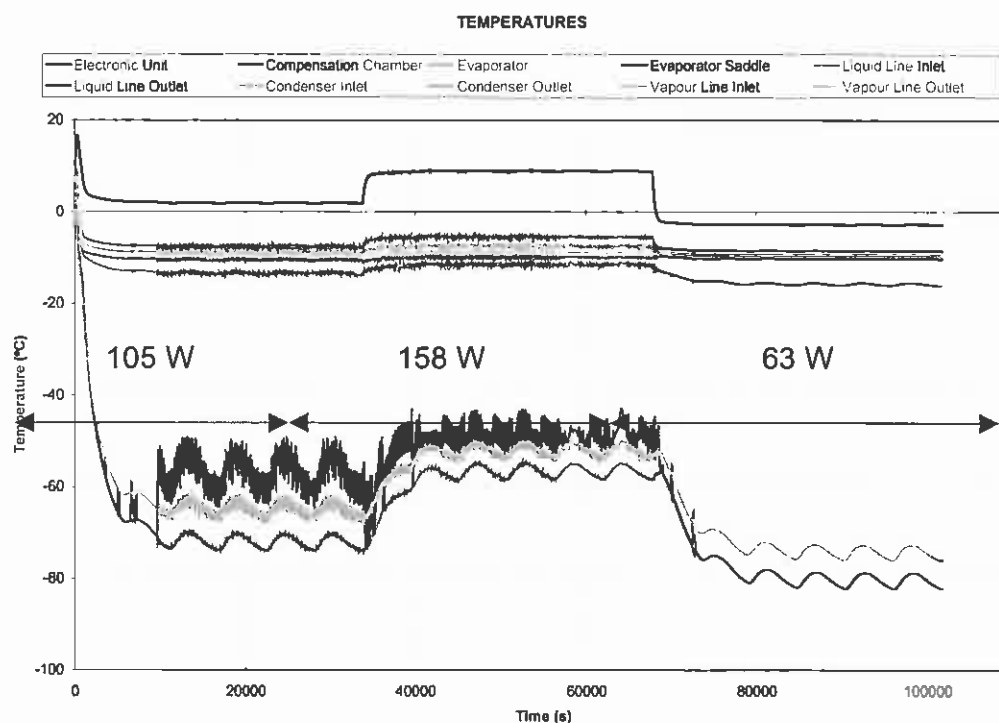


Figure 5-9 : Upper LHP temperatures for the Cold Case

In particular for this case, the temperature of the evaporator saddle in this cold case when applying the minimum power (case which is referred in the requirements as indicated previously) oscillates slightly around -8.5°C , due to the regulation of the valve and is maintained within the specified limits from $+30^{\circ}\text{C}$ and -30°C (required temperature range).

As it is shown in the figure, for the environmental cold conditions, the valve is partially open for the three studied cases and the "ripplets" behaviour is due to numerical imprecision, related to the valve modelling and to the small flow rate. The valve is mass less in the model, and its answer to pressure changes is immediate (no inertia, no spring). This leads to quick oscillations that are expected to disappear in the real behaviour. The real amplitude and frequency of these oscillations (if any) cannot be predicted until test.

The radiator temperatures are represented in the Figure 5-10

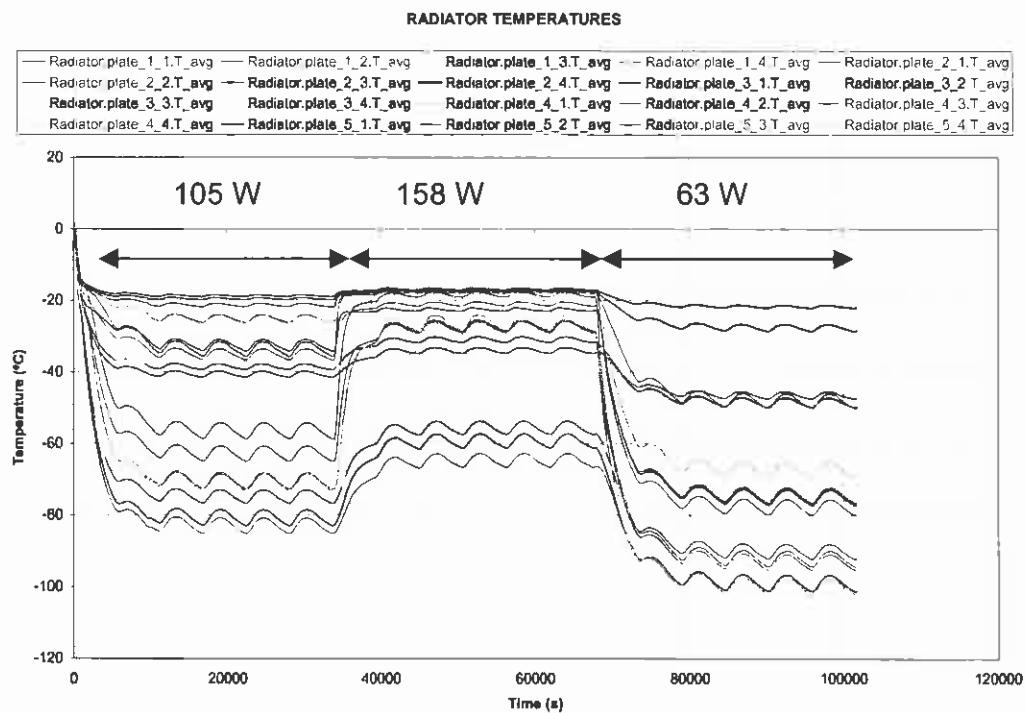


Figure 5-10 : Upper LHP Radiator temperatures for the Cold Case.

The maximum and minimum temperatures calculated for cold conditions are listed in the following table:

Power →	105 W	158 W	63 W
Electronic Unit	$T_{\max} = 2^{\circ}\text{C}$	$T_{\max} = 8.9^{\circ}\text{C}$	$T_{\max} = -2.7^{\circ}\text{C}$
	$T_{\min} = 1.9^{\circ}\text{C}$	$T_{\min} = 8.8^{\circ}\text{C}$	$T_{\min} = -2.9^{\circ}\text{C}$
Evaporator saddle	$T_{\max} = -7^{\circ}\text{C}$	$T_{\max} = -5.2^{\circ}\text{C}$	$T_{\max} = -8.5^{\circ}\text{C}$
	$T_{\min} = -8^{\circ}\text{C}$	$T_{\min} = -5.8^{\circ}\text{C}$	$T_{\min} = -8.6^{\circ}\text{C}$

Table 5-2 : Upper LHP Electronic Unit and Evaporator Saddle maximum and minimum temperatures for the Cold Case

5.4 LOWER LHP

As for the Upper LHP, both LHP systems (RAM and WAKE) are very similar and with no significant differences between, the analysis has been only performed on one of them.

Taking into account these assumptions and those referred in previous paragraphs, we include hereafter the input data for the Lower LHP model:

For reference, the values are taken from Lower RAM LHP System Assy.

5.4.1 Input Data for the Model

The input data are the same that those for the Upper LHP, except:

- COMPONENT – BYPASS LINE

DESCRIPTION	VALUE	UNITS
Total pipe length	0.321	m

- COMPONENT – VAPOUR LINE

DESCRIPTION	VALUE	UNITS
Total pipe length	2.686	m
Radius of curvature of each bend	0.02,0.02,0.015,0.03,0.03,0.02,0.03,0.03,0.015	m
Number of bends	9	
Angle of each bend	108.64,101.69,107.69,120.165.04,165.04,112.3,145.07,90	°

- COMPONENT – LIQUID LINE

DESCRIPTION	VALUE	UNITS
Total pipe length	2.555	m
Radius of curvature of each bend	0.015,0.02,0.02,0.03,0.03,0.03,0.015,0.015	m
Angle of each bend	92.66,91.95,108,120.34,161.93,161.93,86.03,86.03	°
Number of bends	8	-

5.4.2 Scenario

Although the input requirements only refers to max Power for the Hot Case and min Power for the cold case, the model includes, for both cases the next power input profile: first, the nominal dissipation (105 W) is introduced in the Electronic Unit. After six orbital periods, the power is set to the maximum (158 W). Finally, after six orbital periods the power is set to the minimum dissipation (63 W). The power sequence is then : 105 W, 158 W, 63 W.

5.4.3 Results

5.4.3.1 Hot Case

Figure 5-11 represents the temperatures of the Electronic Unit, Compensation Chamber, Evaporator, Transport Lines and the temperatures in the inlet and outlet of the condenser.

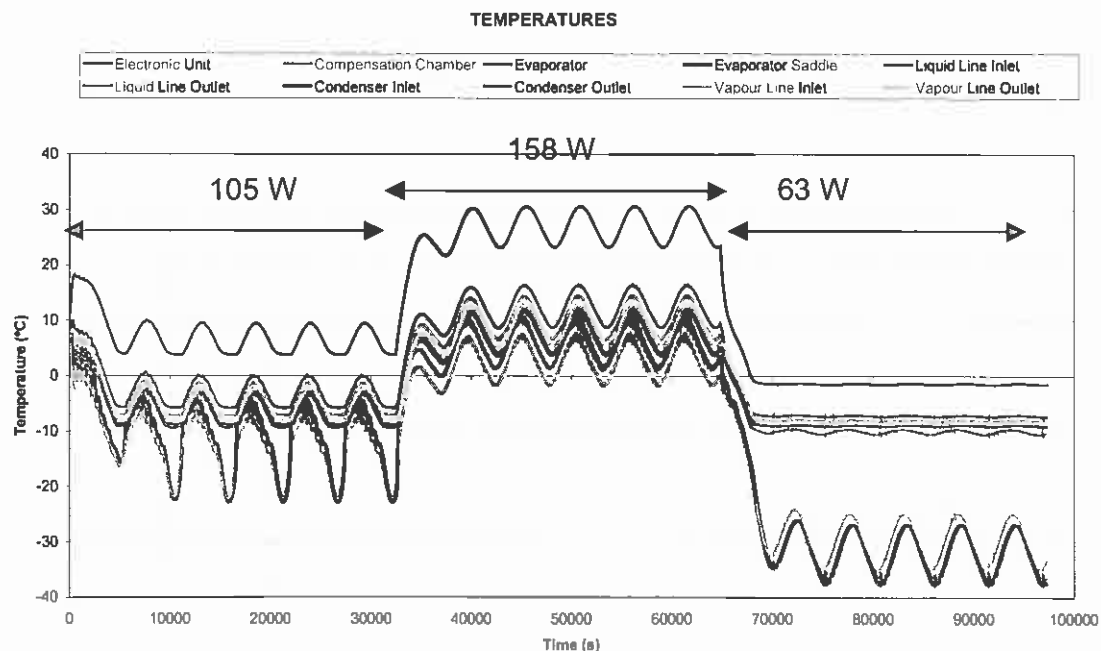


Figure 5-11 : Lower LHP Temperatures for the Hot Case

As showed in Figure 5-11, the temperature of the evaporator saddle is maintained within the required temperature range of -30 and $+30^{\circ}\text{C}$.

In this particular, for this hot case, the temperature of the evaporator saddle when applying the maximum power (case which is referred in the requirements as indicated previously) oscillates between approximately 16 and 9°C and the valve remains closed (all the mass-flow pass through the condenser)

Additionally, regarding the results, we can observe that the valve is regulating the temperature of the evaporator when applying 105 and 63 Watts (the oscillation of the evaporator saddle temperature is flatted). In both cases, the valve is partially open and part of the mass flow goes through the radiator and the other part is by-passed to the Compensation chamber. We can see from the figure that when the by-passed mass flow increases, the decoupling between the "hot side" and the "cold side" are also increased.

Figure 5-12 represents the radiator temperatures in this case.

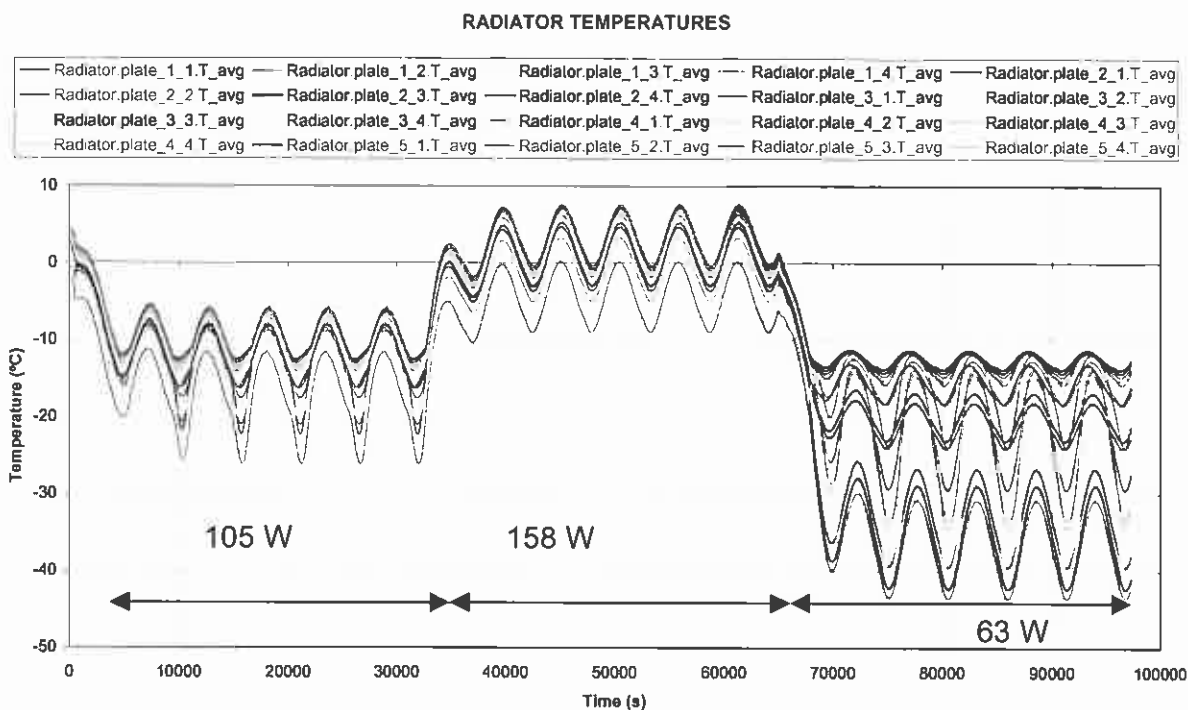


Figure 5-12 : Lower LHP Radiator temperatures for the Hot Case

The maximum and minimum temperatures calculated for cold conditions are listed in the following table:

Power →	105 W	158 W	63 W
Electronic Unit	$T_{max} = 9.4 \text{ }^{\circ}\text{C}$	$T_{max} = 30.3 \text{ }^{\circ}\text{C}$	$T_{max} = -1.3 \text{ }^{\circ}\text{C}$
	$T_{min} = 3.8 \text{ }^{\circ}\text{C}$	$T_{min} = 23.5 \text{ }^{\circ}\text{C}$	$T_{min} = -1.5 \text{ }^{\circ}\text{C}$
Evaporator saddle	$T_{max} = 0 \text{ }^{\circ}\text{C}$	$T_{max} = 16 \text{ }^{\circ}\text{C}$	$T_{max} = -7 \text{ }^{\circ}\text{C}$
	$T_{min} = -5.8 \text{ }^{\circ}\text{C}$	$T_{min} = 9 \text{ }^{\circ}\text{C}$	$T_{min} = -7.3 \text{ }^{\circ}\text{C}$

Table 5-3 : Lower LHP Electronic Unit and Evaporator Saddle approximate maximum and minimum temperatures for the Hot Case

5.4.3.2 Cold Case

Figure 5-13 represents the temperatures of the Electronic Unit, Compensation Chamber, Evaporator, Transport Lines and the temperatures in the inlet and outlet of the condenser.

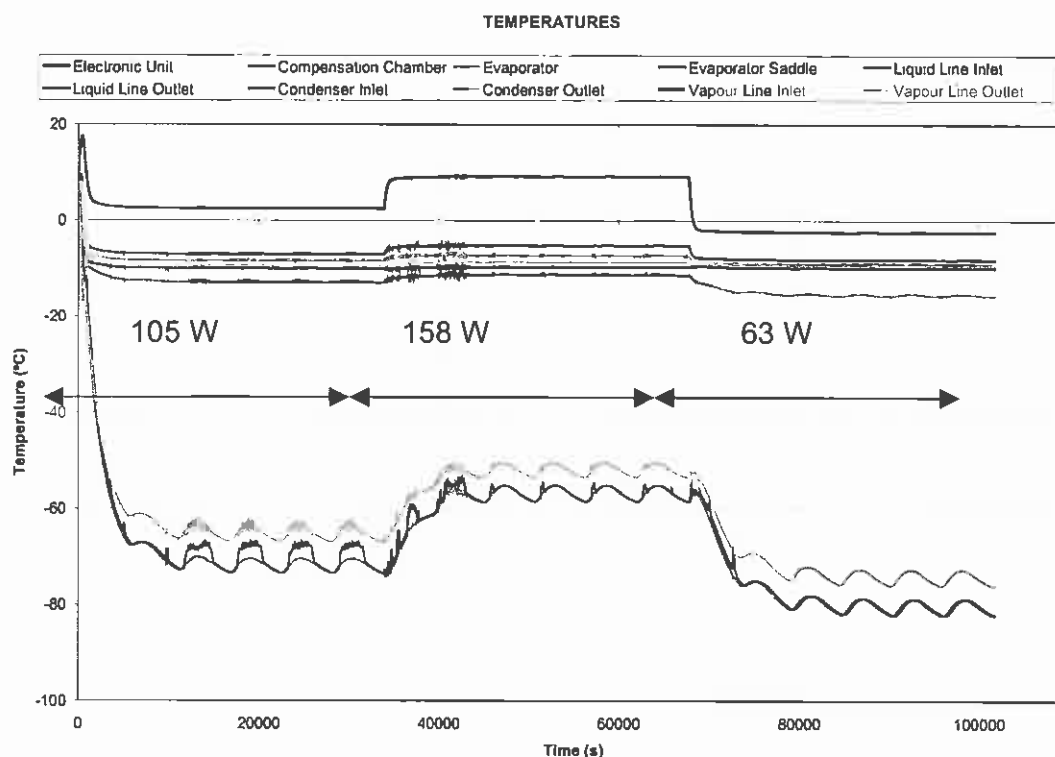


Figure 5-13 : Lower LHP Temperatures for the Cold Case

The temperature of the evaporator saddle in this cold case when applying the minimum power (case which is referred in the requirements as indicated previously) oscillates slightly around -8°C , due to the regulation of the valve. In all the three studied cases, the valve is always regulating the temperature.

As well as for the other cases, the temperature of the evaporator is maintained within the temperature range of $+30^{\circ}\text{C}$ and -30°C (required temperature range).

The radiator temperatures are represented in Figure 5-14

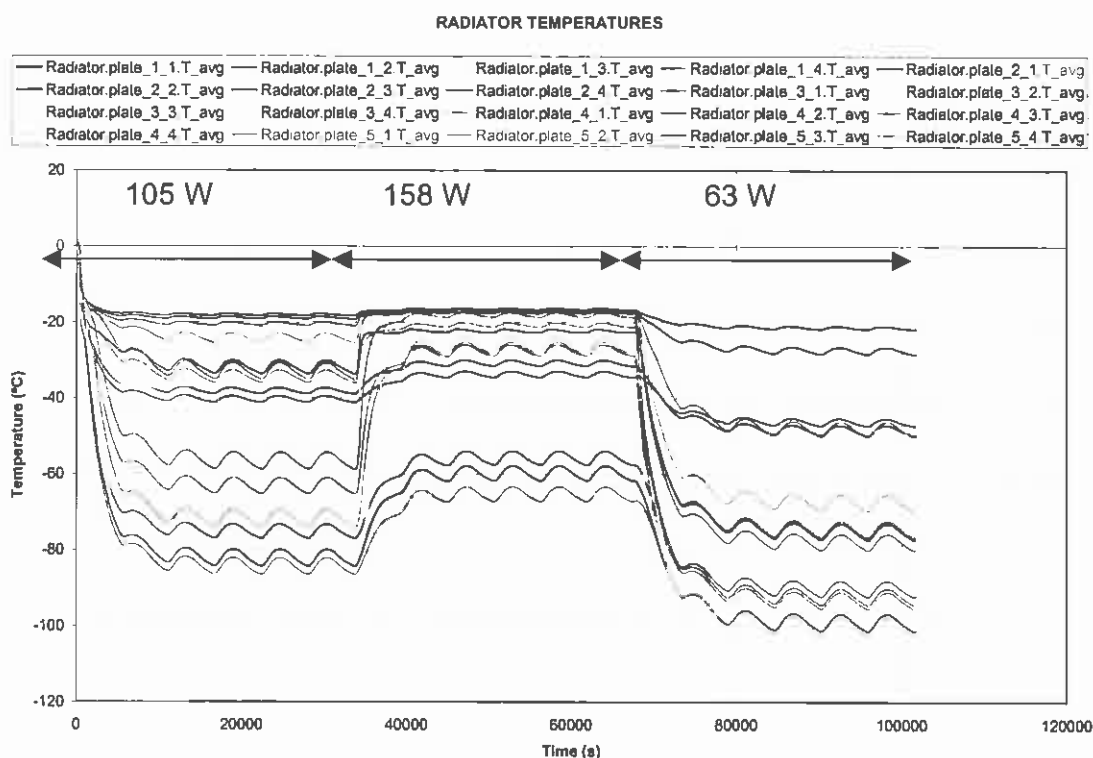


Figure 5-14 : Lower LHP Radiator Temperatures for the Cold Case

The maximum and minimum temperatures calculated for cold conditions are listed in the following table:

Power →	105 W	158 W	63 W
Electronic Unit	$T_{\max} = 2.6^{\circ}\text{C}$	$T_{\max} = 9.2^{\circ}\text{C}$	$T_{\max} = -2.3^{\circ}\text{C}$
	$T_{\min} = 2.5^{\circ}\text{C}$	$T_{\min} = 9.1^{\circ}\text{C}$	$T_{\min} = -2.4^{\circ}\text{C}$
Evaporator saddle	$T_{\max} = -7^{\circ}\text{C}$	$T_{\max} = -5.2^{\circ}\text{C}$	$T_{\max} = -8^{\circ}\text{C}$
	$T_{\min} = -7.1^{\circ}\text{C}$	$T_{\min} = -5.3^{\circ}\text{C}$	$T_{\min} = -8.1^{\circ}\text{C}$

Table 5-4 : Lower LHP Electronic Unit and Evaporator Saddle approximate maximum and minimum temperatures for the Cold Case

6. CONCLUSION

The thermal-hydraulic detail design of the LHP has been presented. For the design, a final model have been developed using the mathematical tool EcosimPro. The main assumptions made in the modelling as well as the results are given in this document. In particular, two cases have been run to verify the fulfilment of the thermal performances in terms of temperature and thermal conductance.

For the hot and cold cases, it has been demonstrated that the operational temperature ranges described in the requirement [AD01] has been respected.

That is, the evaporator saddle temperature is for all the studied cases, within the specified temperature margins (-30 and +30°C)